Heating, Ventilating and Air-conditioning System Energy Demand Coupling with Building Loads for Office Buildings

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Abstract

The UK building stock accounts for about half of all energy consumed in the UK. A large portion of the energy is consumed by nondomestic buildings. Offices and retail are the most energy intensive typologies within the nondomestic building sector, typically accounting for over 50% of the nondomestic buildings‟ total energy consumption. Heating, ventilating and air conditioning (HVAC) systems are the largest energy end use in the nondomestic sector, with energy consumption close to 50% of total energy consumption.

Different HVAC systems have different energy requirements when responding to the same building heating and cooling demands. On the other hand, building heating and cooling demands depend on various parameters such as building fabrics, glazing ratio, building form, occupancy pattern, and many others. HVAC system energy requirements and building energy demands can be determined by mathematical modelling. A widely accepted approach among building professionals is to use building energy simulation tools such as EnergyPlus, IES, DOE2, etc. which can analyse in detail building energy consumption. However, preparing and running simulations in such tools is usually very complicated, time consuming and costly. Their complexity has been identified as the biggest obstacle. Adequate alternatives to complex building energy simulation tools are regression models which can provide results in an easier and faster way.

This research deals with the development of regression models that enable the selection of HVAC systems for office buildings. In addition, the models are able to predict annual heating, cooling and auxiliary energy requirements of different HVAC systems as a function of office building heating and cooling demands.

For the first part of the data set development used for the regression analysis, a data set of office building simulation archetypes was developed. The four most typical built forms (open plan sidelit, cellular sidelit, artificially lit open plan and composite sidelit cellular around artificially lit open plan built form) were coupled with five types of building fabric and three levels of glazing ratio. Furthermore, two measures of

reducing solar heat gains were considered as well as implementation of daylight control. Also, building orientation was included in the analysis. In total 3840 different office buildings were then further coupled with five different HVAC systems: variable air volume system; constant air volume system; fan coil system with dedicated air; chilled ceiling system with embedded pipes, dedicated air and radiator heating; and chilled ceiling system with exposed aluminium panels, dedicated air and radiator heating. The total number of models simulated in EnergyPlus, in order to develop the input database for regression analysis, was 23,040.

The results clearly indicate that it is possible to form a reliable judgement about each different HVAC system‟s heating, cooling and auxiliary energy requirements based only on office building heating and cooling demands. High coefficients of determination of the proposed regression models show that HVAC system requirements can be predicted with high accuracy. The lowest coefficient of determination among cooling regression models was 0.94 in the case of the CAV system. HVAC system heating energy requirement regression models had a coefficient of determination above 0.96. The auxiliary energy requirement models had a coefficient of determination above 0.95, except in the case of chilled ceiling systems where the coefficient of determination was around 0.87.

This research demonstrates that simplified regression models can be used to provide design decisions for the office building HVAC systems studied. Such models allow more rapid determination of HVAC systems energy requirements without the need for time-consuming (hence expensive) reconfigurations and runs of the simulation program.

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This work has been made possible by the help, encouragement and support from my supervisors, colleagues, family and friends.

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I would also like to thank my friends and former colleagues from the CityNet project and at the Institute of Energy and Sustainable Development (IESD) where I did my research. An incomplete list includes Meltem Bayraktar, Alberto Coronas, Eric Duminil, Ursula Eicker, Sherif Ezzeldin, Denis Fan, Donal Finn, Julie Futcher, Miaomiao He, David Infield, Katherine Irvine, Dilay Kesten, Jerko Labus, Rob Liddiard, Kevin Lomas, John Mardaljevic, Divine Novieto, Marco Perino, Tobias Schulze, Rafal Strzalka, Graeme Stuart, Aysegul Tereci, Roman Ulbrich, Maria Carla Di Vincenzo, Andrew Wallace, Zerrin Yilmaz (sorry if I have missed anyone).

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Chapter 1

Introduction

In the age of rapid anthropogenic global climate change and carbon emission reduction targets, the energy efficiency of urban areas has been identified as a key factor. Ever since 1993, various EU documents were clearly indicating the importance of the energy reduction in the building sector, often identifying it as having the largest cost-effective saving potential. In the UK, buildings account for about half of all energy, compared to 41% in Europe and 36% in USA (Steemers, 2003). A reduction of CO_2 emissions in urban areas of only 10% would have a big impact on greenhouse gas emissions. In cities, there is a concentration of non-domestic buildings so they are likely to account for a disproportionate amount of energy use.

Offices and retail are the most energy intensive typologies within non-domestic building sector, typically accounting for over 50% of the total energy consumption for non-domestic buildings (Pérez-Lombard et al., 2008). Especially important has been the intensification of energy consumption in heating, ventilating and air conditioning (HVAC) systems which have now become almost essential in parallel to the spread in the demand for thermal comfort. It is the largest energy end use both in the residential and non-residential sector with a weight close to 50% in non-domestic buildings. According to Pérez-Lombard et al. (2008) it is advisable to start the analysis of energy demand of the non-domestic building stock with office buildings. The reason is not only the energy intensity of the office buildings but their constant increase in total floor area coupled with increase in lighting, IT and air-conditioning. The other important reason is that office buildings are quite uniformly distributed across the buildings stocks in developed countries with three key energy end-uses, HVAC, lighting and appliances, adding up together to around 85%.

When trying to address the specific non-domestic typology, like office buildings for example, the usual approach is by defining a typical or exemplar office building. That approach has been used by Jenkins et al. (2008, 2009) to investigate the refurbishment options aiming at 50% reduction in CO2 emission inside UK office building stock. The sensitivity of future UK office buildings energy demand and subsequent consumption to future climate changes and improvements in HVAC and appliance and lighting efficiency are investigated based on an example building whose shape represents 20% of the office building stock and construction age represents 9% of the office building stock. Office building energy consumption was determined for a generic HVAC system described by its boiler and chiller efficiencies for both design and part load conditions. Dascalaki and Santamouris (2002) have reported the results from the OFFICE project in which European office buildings were classified into five typical types in order to investigate passive and active measures to reduce CO_2 emission in five different climatic zones. One of the reported finding is that the type of the HVAC system significantly influences total energy consumption.

Modern buildings and their HVAC systems are nowadays required not only to be more energy efficient while adhering to an ever-increasing demand for better performance in terms of comfort, but equally in respect to financial and environmental issues. Managing adequately the air-conditioned building energy demand has always been a struggle for facility managers (Neto and Fiorelli, 2008). The choice of HVAC systems impacts the life-cycle cost of the building; a building with an ineffective HVAC system or high running cost is also unlikely to be leased or sold easily (Ellis and Mathews, 2002). It is not surprising that the design of comfortable, energy efficient buildings is receiving a lot of attention. Research in this field tends to focus on computer software applications aimed at reducing energy consumption. Historically the emphasis was on building energy demand rather than actually energy consumption. However, with the ever increasing demand for cooling even in mild climates such as UK (Caeiro et al., 2008) the research into HVAC system selection is becoming more important.

The overall environmental impact of any building in terms of Carbon emission depends on the amount of fuel consumed by its HVAC system and the fuel type. Energy flow of principal HVAC system within buildings is presented in Figure 1.1. HVAC system is usually divided into two parts; primary HVAC system and secondary HVAC system. Primary HVAC system is composed of equipment which generates heating/cooling energy (Q_h, Q_c) from primary fuels. Typical examples of primary

HVAC system equipment are boilers and chillers which operate with certain efficiency (η) and coefficient of performance (COP). Heating/cooling energy is distributed through a building by a secondary HVAC system in order to respond to a building heating/cooling demand. Secondary HVAC system requires additional energy, so-called auxiliary energy (Q_a), to operate mechanical components of a system such as pumps, fans and control gears.

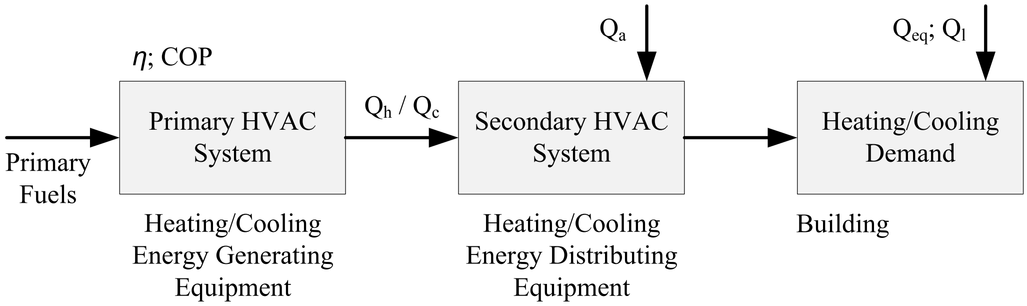


Figure 1.1. Energy flow of principal HVAC system within buildings

Building heating/cooling demand is the amount of heating/cooling energy required to maintain desired indoor conditions. Building demands are usually calculated by taking into consideration only typical heat gains and heat losses which occur in buildings which are: transmission heat gains/losses through building envelope elements, solar heat gains through fenestration areas, internal heat gains from occupant, artificial lighting and electrical equipment, infiltration air heat gains/losses, and fresh air ventilation heat gains/losses.

Building heating/cooling demand depends on various building parameters such as building fabrics, glazing percentage and glazing properties, occupancy pattern, level of internal gains, etc. Building demand calculation is often used in studies where the main task is to investigate the performance of building and its components. In addition, it is very useful in comparison studies for exploring different design options and analysing the influence of various building parameters, individually or in combination, on building thermal behaviour. Despite that the building demand calculation is often used for building‟s energy performance evaluation in the practice, it is not very useful for determining energy requirements of buildings serviced by an HVAC system. The amount of energy the HVAC system demands from the primary sources, in order to deliver required heating and cooling to the building, does not equal the building heating/cooling demand in most circumstances. Different HVAC systems have different energy requirements when responding to the same building heating/cooling demand. Such behaviour is predominantly affected by the way a particular HVAC system is designed and operated. For example, systems which cover building demands by distributing only the air to the zone with energy requirements (all-air systems) can benefit from free cooling by using an air-side economizer. On the other hand, systems which operate with the constant amount of air, enough only to cover fresh air requirements, can benefit from the heat recovery unit, which should affect both heating and cooling energy consumption.

Building demand and energy requirements of HVAC systems can be determined by mathematical modelling. There are two modelling approaches suitable for applying in building services science: forward (classical) approach and data-driven (backward) approach. These two approaches are explained in the ASHRAE Handbook of Fundamentals Chapter 19 (ASHRAE, 2009). Forward models are usually very complex and require large numbers of input parameters to be specified in order to calculate the desired outputs. Some of the required input parameters are, for example, detailed building geometry, location, thermo-physical characteristics of building construction elements, type of HVAC system, control system, operating schedules, and many others. Forward models are widely accepted by the building professional community and most of the software packages developed to predict building energy use are based on this approach such as EnergyPlus, BLAST, DOE-2, IES to mention just a few. Although computer design tools have a tremendous potential for aiding designers and other professionals in the built environment in achieving high standards of building energy efficiency, the complexity of the existing tools has been identified by Ellis and Mathews (2002) as their biggest obstacle. The same research suggested that thermal efficiency of buildings and the selection on HVAC systems are two areas that can benefit from simplified tools which will simplify input complexity by identifying and focusing on critical parameters and defining them in architectural terms. In achieving these, the simpler tools might be more appropriate for the wide spread use by professionals in built environment especially at the initial design stage when the majority of the energy intensive decisions are made.

The data-driven approach typically uses known input and output variables to generate a mathematical description of the building or HVAC system as a function of various influential variables such as outdoor temperature, solar radiation, HVAC systems characteristics, etc. Data-driven models are usually single or multiple regression models and they are much simpler than forward models. In addition to simplified regression models, the more complex models, such as artificial neural network models and Fourier series models, can also be created by data-driven modelling approach.

Data-driven modelling is not widely accepted by building professionals despite that there are quite a few research studies which results confirm usefulness of such approach. Sander et al. (1993) developed simplified regression models which predict building annual heating and cooling energy requirements for a building equipped with a generic variable air volume (VAV) air-conditioning system based on location, building envelope characteristics and internal gains. The outputs form 5400 building simulation for 25 Canadian locations were used as regression analysis inputs. The accuracy of developed models was quite high with a difference between model predictions and simulation outputs within 10% in most cases, except for buildings with very low either heating or cooling requirements.

A similar approach was used in obtaining the equation which can predict annual energy consumption of high-rise fully air-conditioned office buildings in Hong Kong (Lam et al., 1997, Hui, 1997). A generic office building was simulated in the DOE-2 energy simulation software by varying 62 input design parameters related to the building demand, HVAC system and HVAC refrigeration plant. Authors reported that from 62 input parameters, 28 correlate well with the predicted annual energy consumption. After performing a sensitivity analysis, 12 of 28 input parameters were considered to have the most significant impact on energy consumption and they were used in the regression analysis. A regression model based on the 12 parameters was able to predict building annual energy consumption with a high accuracy, having the coefficient of determination close to 0.99. Lam et al. (2010) extended this study by including an additional four climate regions in China. The new model was also based on 12 characteristic parameters, although parameters were slightly changed in order to include the impact of heating on building energy consumption. New equations (one equation per climate region) were also capable to predict building annual energy consumption with a coefficient of determination between 0.89 and 0.97, depending on the climate.

Bansal and Bhattacharya (2009) also used the detailed simulation results for developing simplified equations which can predict a single zone building annual energy demand as well as maximum heating and cooling loads for the central India weather conditions. Equations were presented as a function of either the insulation thickness or the surface to volume ratio. In the case of the surface to volume ratio, the impact of increasing the floor-to-roof height and varying both length and depth was investigated. Simplified equations presented in the paper provided a very good fit with a lowest coefficient of determination above 0.9. Ourghi et al. (2007) presented a simplified method, based on regression analysis, for office building annual cooling and total energy use in Kuwait and Tunis. Exemplar office buildings of various shapes were modelled using DOE-2 with typical office occupancy patterns and schedules with the same HVAC system in all cases: VAV with electric re-heat for both heating and cooling. The research found a strong correlation between annual total energy use and building relative compactness (“cube-likeness”), window to wall ratio and the glazing solar heat gain coefficient for cooling dominated climates. Jaffal et al. (2009) developed simple polynomial functions which predict the annual energy demand as a function of building envelope parameters for both cold and moderate climate in France. Polynomial functions were based on the 11 most influential envelope parameters, which can be chosen in the design stage, and on environmental inputs. According to authors, the advantage of polynomial functions lies in simplicity, speed and precision in evaluating the energy saving potentials of various building elements, individual or in combination, for which dynamic simulation would be very time consuming.

Abovementioned researches confirm that regression equation models are adequate alternative to complex building simulation tools for predicting HVAC system energy consumption and building demands, particularly during the early design stage when different designs need to be explored and evaluated. Regression equation models can provide precise and accurate results in an easier and faster way than building energy simulation tools, especially if they are developed from a comprehensive, broad and accurate input dataset.

1.1. Research aim and objectives

The aim of this research is to contribute towards advances in the development of the simplified, yet accurate, models which will enable the selection of the heating, ventilating and air-conditioning (HVAC) systems for office buildings. The models will be able to predict annual heating, cooling and auxiliary energy requirements of different secondary HVAC systems as a function of office building heating and cooling demands.

In order to develop models which determine energy requirements of different secondary HVAC systems, it was assumed that the primary system operates with 100% efficiency and it is capable of providing enough energy at desired temperature to fulfil all requirements all the time. In addition, since the function of HVAC system is to provide and maintain satisfactory indoor environment conditions, it is important to mention that the only indoor air parameter which is precisely controlled is the air dry-bulb temperature and that humidity control is excluded from the study.

In developed models, building heating and cooling demands were selected as independent variables since they can be considered to be a building characteristic. Different HVAC systems will have different energy requirements, when coupled with a building with particular heating and cooling demands. In addition, setting up and running a building demand calculation in detailed building energy simulation tools is less complex and requires less time than setting up a simultaneous simulation of building and HVAC system. Extending building simulation with an HVAC system adds extra levels of complexity since there are additional parameters related to the system which need to be specified, such as HVAC system controls, temperature regimes, operating schedules, auxiliary equipment characteristics, primary equipment characteristic, etc.

The existence of such simplified models, which can predict secondary HVAC systems heating, cooling and auxiliary energy requirements as a function of building heating and cooling demands, may be useful in several situations. For example, during early design stage, when energy intensive decisions are made, simplified models can save time and provide fast and accurate way to explore different HVAC systems and their impact on a building energy end-use and greenhouse gas emissions. In addition, in refurbishment projects such tools can help in making a decision about replacing an existing HVAC system with another one which will have reduced energy consumption. Another possible area of use is energy end use modelling of non-domestic building stocks, in particular office building stock, at Community level where Community can be anything from part of the city up to national level.

In order to achieve this aim the following objectives were determined:

- to describe the non-domestic building stock in the UK with a particular emphasis on office buildings,
- to explore through literature review which building characteristics are important for building energy consumption (for example, building shape and form, orientation, glazing ratio, construction elements characteristics, space arrangement, etc.),
- to develop office buildings" simulation models archetypes which will represent the current and future UK office building stock,
- to evaluate the influence of different HVAC systems on building energy end use through comprehensive literature review,
- to develop detailed simulation models of the most popular HVAC systems in the UK,
- to run building energy simulation of office buildings coupled with different HVAC systems as well as building demand calculations in order to prepare the dataset for the regression analysis,
- to apply regression analysis to the dataset and to statistically examine the outputs of the regression analysis in order to recommend suitable simplified models which can predict annual heating, cooling and auxiliary energy requirements of different HVAC systems as a function of building heating and cooling demands.

1.2. Research methodology

In order to create simplified HVAC system heating, cooling and auxiliary energy requirements models both forward (classical) and data-driven (backward) modelling approaches were used. Firstly, the dataset used in regression analysis were generated by using the forward approach. Detailed simulations of buildings equipped with different HVAC systems were performed in EnergyPlus and the annual energy requirements, normalised per square meter, were calculated and stored in the database. The backward approach was then applied to this database to generate a mathematical description of the HVAC system energy consumption as a function of building heating and cooling demands. After that, the statistical analysis on the various models developed was performed in order to determine the most suitable set of models which fulfil the two most important requirements: to be accurate and to be simple.

1.3. Thesis structure

The thesis is composed of five chapters and three appendixes. The current chapter presents the aim and objectives of the research study and provides the scientific background about the foundation of this research.

Chapter 2 describes the development of the UK office buildings‟ simulation archetypes for both existing and future stock. Chapter 2 firstly presents the classification of the UK non-domestic building stock in particular focusing on office buildings. The classification is followed by the literature review on office building built form, building construction parameters such as insulation levels, glazing percentages, infiltration, etc. and building parameters related to environmental conditions and building activity which include lighting, appliances and occupancy density. The last part of the chapter gives the rationale behind office building simulation archetypes parameters selection. The selected built forms, insulation levels, glazing percentages, indoor space layouts were combined into a large set of office building models (3,840 in total).

Chapter 3 presents the classification of HVAC systems and briefly describes basic characteristics of various HVAC system types. The systems description is followed by a comprehensive literature review on the energy efficiency of different

HVAC systems and the systems distribution across an office building stock. Last part of the chapter is focused on the detailed description of HVAC system simulation models developed for the purpose of this research.

Chapter 4 firstly describes the effective way of running a large number of simulations with a limited number of input files. After that, it presents the analysis of HVAC systems simulation outputs. The analysis was conducted in two parts: the first part shows the analysis of office buildings energy requirements when coupled with different HVAC systems; the second part presents the regression analysis which resulted in generating mathematical models which can predict cooling, heating and auxiliary energy consumptions of particular HVAC system as a function of building demands. Chapter 4 is concluded with a summary of major findings from the regression analysis as well as with the possible constraints of the models developed.

Chapter 5 presents the conclusion and possible directions of further work which can be based on findings from this research.

Appendix A describes the physical properties and configuration of the building envelope and building interior elements.

Appendix B presents the detailed literature survey on validation of EnergyPlus building simulation software.

Appendix C consists of tables and figures which were generated by the regression analysis.

Chapter 2

Office Buildings

2.1. Non-domestic buildings

Non-domestic building stock, which is quite heterogeneous, is composed of buildings for all uses other than houses and flats and it can be classified in many ways. One quite common way of classification is by the property ownership into private sector and public sector. The private sector includes commercial offices, shops, factories, warehouses, hotels, catering establishments, communication establishments, etc. In the public sector, there are central and local government offices, educational institutions (schools, colleges, and universities) and defence establishments. Some building types, such as health sector premises (surgeries, clinics, hospitals, etc.), overlap across public and private sector.

The UK non-domestic building stock can be explored by using data sets from the Valuation Office Agency (VOA). The VOA holds data, such as floorspace and rateable value statistics, on every property in England and Wales which is subject to commercial rates. The data are organised by „hereditaments" which are defined, according to the Office of the Deputy Prime Minister document (ODPM, 2006c), as property on which rates may be charged. In practice, a hereditament may be a room or part of the floor in the building, a whole building or a group of buildings. Some are not buildings at all and consist only of land, for example, open-air car parks, storage land, and various kinds of sports grounds. Hereditaments are grouped into bulk classes. The bulk class properties are those for which floorspace and other descriptive information is consistently available. There are five bulk classes defined:

- Retail premises,
- Offices,
- Factories,
- Warehouses, and
- Other premises.

Retail premises are premises which provide „off-street" goods and services to the public and typical examples are supermarkets, corner shops, local post offices, bank branches, restaurants, etc. The offices bulk class is composed of hereditaments which serve mainly commercial activities such as purpose-built office buildings, offices over shops, light storage facilities and light industrial activities. This class also includes larger banks, building societies and post offices which contain substantial office space. The factories bulk class ranges from small workshops to very large manufacturing units, while warehouses range from small storage units to very large distribution warehouses. Car showrooms are also included in the warehouses bulk class. The other premises bulk class includes mainly „community" type establishments such as community centres, village halls and social clubs. Properties which do not fall into one of the five bulk classes and generally do not have floorspace and other descriptive statistics available are collectively known as the „non-bulks" and include car parks, pubs and schools (ODPM, 2006c).

The document produced by the Department for Communities and Local Government in collaboration with the VOA provides a summary of hereditament, floorspace and rateable value statistics for non-domestic property in England and Wales as at 1st of April 2008 (DCLG, 2009). Figure 2.1 shows number and percentage of hereditaments in each bulk class. It can be seen that the retail bulk class has the highest number of hereditaments, nearly 550,000 units which is 30.5% of the total number of hereditaments whilst they are slightly above 350,000 office properties equalling to 19.5% of the total number of hereditaments. However, in terms of the total floor distribution (Figure 2.2), the highest total floor area is occupied by factories followed by warehouses. The retail bulk class and the office bulk class follow with similar total floor areas, 106.3 and 101.5 millions square meters respectively.

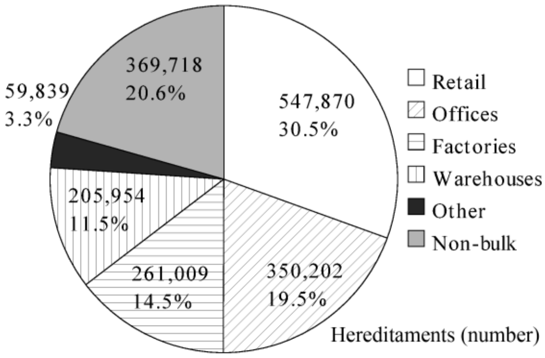


Figure 2.1. Commercial and industrial property: hereditaments - summary statistics England and Wales, 1st April, 2008

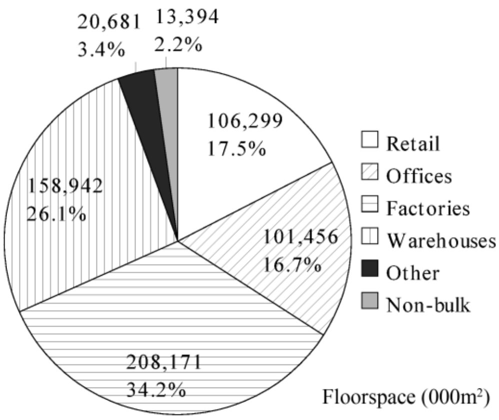


Figure 2.2. Commercial and industrial property: floorspace - summary statistics England and Wales, 1st April, 2008

This research is focused on office buildings only. The office building can be defined as a place in which business, clerical, or professional activities are conducted. The VOA subdivided the office bulk class into two classes called „commercial" offices and non-commercial „other" offices. The commercial offices group is composed mainly of purpose built office buildings and various types of non-domestic buildings converted to offices, offices over shops and computer centres. Central government offices are also included in this category. The „other" office category includes mainly local government offices, surgeries and clinics, and police stations. Both figures, Figure 2.3 and Figure 2.4, show the ratio between the commercial office class and the „other" office class for the number of hereditaments and the total floorspace respectively. „Other" offices are less common and account for 14.3% of the number of hereditaments and 17.1% of the total floorspace in the office bulk class.

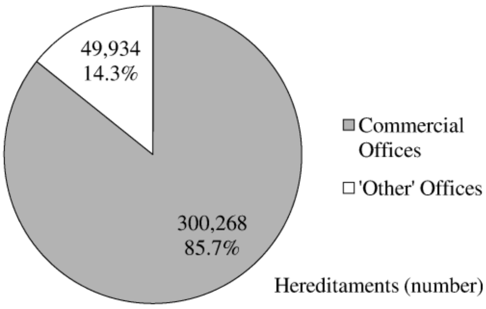


Figure 2.3. Commercial and 'other' offices: hereditaments - summary statistics England and Wales, 1st April, 2008

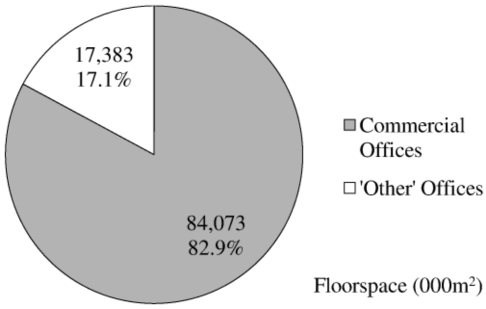


Figure 2.4. Commercial and 'other' offices: floorspace - summary statistics England and Wales, 1st April, 2008

Besides the share of office buildings in the non-domestic building stock and the ratio between commercial and „other" offices, it is interesting to see the age profile of the office building stock. The Office of the Deputy Prime Minister published the document „Age of Commercial and Industrial Stock: Local Authority Level 2004" (ODPM, 2005) which provides analysis of the age of non-domestic properties as at 1st April 2004. The document is also based on the data sourced from the VOA. Figure 2.5 shows the age distribution of office properties in England and Wales. Just over half of office hereditaments were built before 1940 while properties completed between 1990 and 2003 account for 12% of the total stock. However, Figure 2.6 shows that total office floorspace area is much more evenly distributed between the different age ranges. Just below 28% of all offices were built before 1940 while a 20% were constructed between 1990 and 2003. By comparing Figure 2.5 and Figure 2.6, it can be concluded that newer properties tend to have a larger floorspace on average. In addition, it can be observed that a certain number of properties are of unknown age. There is no year built information for about 1.7% of office hereditaments which accounts for around 5.8% of the total floorspace.

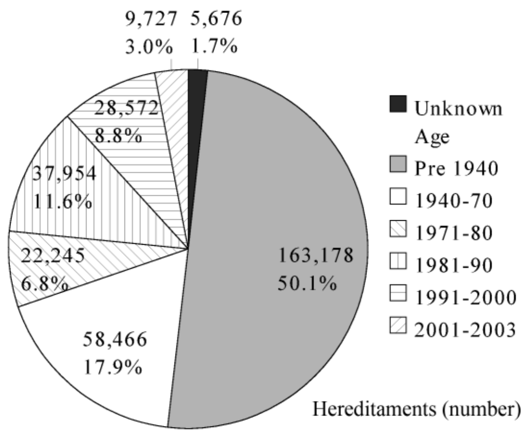


Figure 2.5. Age profile of offices: hereditaments - summary statistics England and Wales, 1st April, 2004

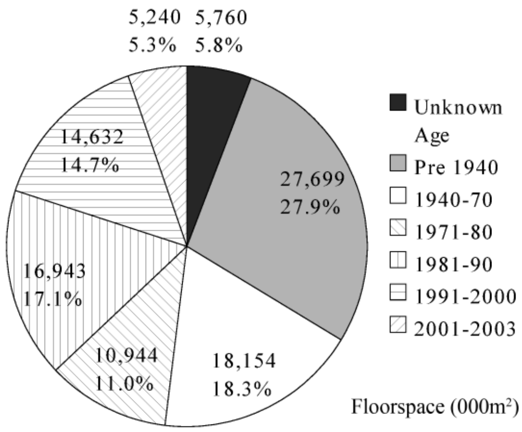


Figure 2.6. Age profile of offices: floorspace - summary statistics England and Wales, 1st April, 2004

2.2. Office building built forms

Building geometry can have a significant impact on building energy demand and, therefore, special attention has to be paid to a classification of built forms. A detailed study of building geometries resulted in the identification of several principal built forms. The study was based on data from a series of local surveys carried out in four English towns in which the non-domestic building stocks might be considered characteristic of the national stock: Manchester, Swindon (Wiltshire), Tamworth (Staffordshire) and Bury St Edmunds (Suffolk). These four towns were selected for their range of population size, widely spread geographically across the country, a great variety of building types, and the fact that none is dominated by a single industry (Brown et al., 2000). In each case, a sector of the town was chosen and all non-domestic buildings in that area were surveyed. The surveys were part of a project to develop a national Non-Domestic Building Stock (NDBS) database which main purpose was to provide a better statistical picture of the non-domestic stock, and of uses of energy in non-domestic buildings (Steadman et al., 2000a, Steadman et al., 2000b). In total, some 3,350 addresses were covered by survey with the total area of floorspace just under 4 million m^2 (gross external area). Each building was inspected externally and a large number of building characteristics were recorded. The recorded data included: building estimated age, overall building form (including roof type and number of storeys), and details of fabrics and constructions visible from the outside such as structural type, glazing type, and external wall and roof finishing.

The classification of built forms was made according to two basic criteria: whether or not a space is predominantly daylit or artificially lit and a space layout (Steadman et al., 2000a). Authors observed that rooms take typical ranges of size depending on their functions and created three subcategories: cellular spaces, open plan spaces, and halls. Cellular space arrangement is typical for strings of individual offices in commercial buildings, bedrooms in hotels, classrooms in schools, etc. Such rooms are more or less comparable sizes, equipped and furnished in similar ways, serve quite standardised purposes, and accommodate roughly equal number of occupants. Open plan spaces and halls are similar in both size and shapes, and both are unobstructed by walls. The difference between them is in the occupants" activity. Halls are large single specialised spaces occupied by single coordinated activity such as lecture theatres, conference and meeting rooms, assembly halls, churches and chapels, cinemas, etc. On the other side, the occupants of the open plan spaces are engaged in many different activities. In most cases, this type of space accommodates office activities, but the same description can be applied to many large shops or warehouses. By combining these two basic criteria, Steadman et al. (2000a) proposed the six basic built forms:

- Daylit cellular,
- Artificially lit cellular,
- Daylit hall,
- Artificially lit hall,
- Daylit open plan, and
- Artificially lit open plan.

The daylit cellular group includes most of office accommodation, except open plan offices, and majority of the space in hotels. The artificially lit cellular built form is quite rare and occurs mainly in spaces such as basements. Large spaces such as churches, which are generally daylit from the side, or courtyards, which most of them are daylit from the top, are typical examples of the daylit halls. Cinemas, theatres or television studios fall into the artificially lit halls category. Typical examples of the daylit open plan category are spaces which are toplit, for example single-story buildings or top floors, or sidelit spaces with maximum depth which allows an acceptable level of daylighting. Artificially lit open plan category includes all types of large spaces with none or very limited daylighting such as large open plan offices, majority of shops, and some of warehouses.

Beyond this, Steadman et al. (2000a) adopted several strategies to simplify the representation of complex built form. All minor details of form are ignored, such as building attachments, balconies, small bay windows, etc. Buildings of complicated form are disaggregated into smaller component parts, and, where is suitable, these separate built form components are assigned to different classification categories. Some building forms are subcategorised according to numbers of storeys to make a distinction between forms in which vertical communication can be obtained by staircase only (forms up to four storeys) and forms for which lifts are necessary (forms with five storeys and more). Schematic drawings of the principal built forms are given in Figure 2.7. The sidelit strip built forms are illustrated with constant depth and straight lengths. However, actual buildings might consist of multiple strips, varying depth and can even have curved plan. Artificially lit built forms are represented in Figure 2.7 with simple rectangular plans, although they have no limitations on the building depth and could have many various plan shapes.

Key

CS4	Cellular daylit strip (1 to 4 storeys)
CS5	Cellular daylit strip (5 storeys & above)
OD4	Open plan daylit strip (1 to 4 storeys)
OD5	Open plan daylit strip (5 storeys & above)
CT1	Cellular toplit single storey
HD	Daylit hall, sidelit, toplit or both
HA	Artificially lit hall
OS	Open plan space in single shed
OC1	Open plan continuous single-storey space
OG	Open plan car parking or trucking deck
OA	Open plan multi-storey artificially lit space
SR	Single-room form
SSR	String of single-room forms
RA	Railway arch
CDO	Daylit cellular strip around some or all edges of artificially lit or toplit open-plan space

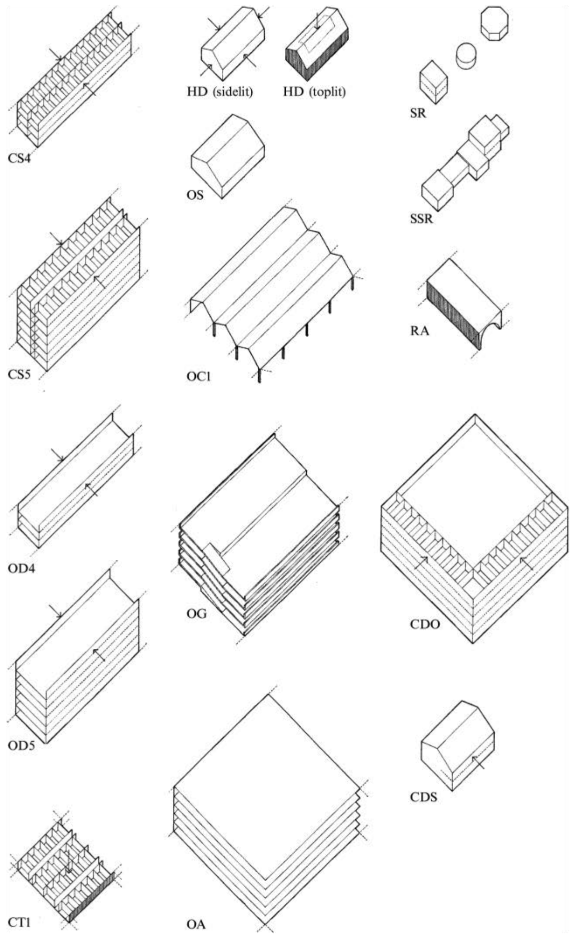


Figure 2.7. Diagrams of principal forms (Steadman et al., 2000a)

All floor space from the surveyed addresses in the four towns were categorised according to this classification. Steadman et al. (2000a) gives the total gross area of floor space devoted to each type of principal form listed in the classification. However, only built form categories which are suitable to accommodate office activity, beside other activities, are presented in Table 2.1 individually. Other built form categories are grouped either into „other principal forms" if they belong to the principal built form category or into parasitic forms. Parasitic form is any form which accounts for significant amount of floor space, especially if that floor space is heated or otherwise serviced, and cannot be added to any of six basic categories of space. Typical examples of parasitic forms are atrium, basement, large balcony, circulation bridge, attached circulation tower, small single-story extension, occupied pitched roof or attic, roof-level plant room, etc.

Table 2.1 shows that the sidelit cellular up to four storeys tall (CS4) is the most common built type with nearly 34% of the floor area. Five % is the share of the sidelit cellular more than four storeys tall (CS5) built type. The sidelit cellular built form category dominates the non-domestic building stock with roughly 40% of the total gross floor area. Second largest individual built form category is the composite sidelit cellular around artificially lit open-plan (CDO) which occupies approximately 20% of the total floor area. The majority of the corresponding activities in these two categories are offices, shops and hotels. Open-plan artificially lit multi-storey space (OA), which accounts for 5.5% of the total, is used for warehouses, factories, offices and shops. The sidelit open-plan up to four storeys (OD4) and the sidelit open-plan with five storeys or more (OD5) use together nearly 3% of the floor area and accommodate mainly offices.

Table 2.1. Total gross floor areas in principal form types in the surveyed buildings

Type	Floor area [m^2]	Percentage
CS4	1 343 247	33.9
CS5	207 516	5.2
CS4 + CS5	1 550 763	39.1
OD4	36 615	0.9
OD5	79 632	2.0
OD4 + OD5	116 247	2.9
CDO	771 402	19.5
OA	216 322	5.5
Other principal forms	871 808	22.1
Parasitic forms	432 631	10.9
Total	**3 959 173**	**100**

2.3. Office building parameters other than built form

Buildings are complex structures and have a lot more parameters, beside a shape and floor layout, which affect its energy consumption. Parameters can be grouped into two categories. The first category includes building construction parameters such as insulation levels, glazing percentages, infiltration, etc. The second category is related to environmental conditions and building activity and includes lighting, appliances and occupancy density.

2.3.1. Building envelope parameters

Building envelope consists of building components which separate the interior of a building and the exterior environment. Most important components are external walls, roof, foundation and fenestration. A fenestration is any area on the exterior building envelope which allows daylit penetration and includes windows, glass doors, and skylights.

By acting as a thermal barrier, a building envelope plays an important role in regulating interior conditions, as well as in determining a building‟s energy use. A building‟s energy use can be affected by a building envelope in many ways and one of the key factors in reducing energy needs is to minimize heat transfer through a building envelope. As a result, special attention during building design is always paid to the insulation of roof and walls, appropriate selection of glazing and framing for windows, airtightness of the building fabrics, suitable shading strategy, etc.

2.3.1.1. Insulation

There is no doubt that increasing the level of insulation results in the significant reduction in heating energy consumption. Aksoy and Inalli (2006) compared buildings without insulation and with 50 mm insulation material and calculated that heating energy savings are around 35% regardless of the building shape and orientation. Several studies optimized the insulation thickness as a function of different energy sources and life-cycle costs. Dombayci et al. (2006) calculated the optimum insulation thickness of the external wall for the five different energy-sources (coal, natural gas, LPG, fuel oil and electricity) and two different insulation materials (expanded polystyrene and rock wool) for the city of Denzili in Turkey. With a use of optimum insulation thickness, heating energy consumption decreased by 46.6% (Dombayci, 2007). Bolattürk (2006) performed similar calculation by including 16 cities from four climate zones. He calculated that depending on the city and the type of fuel, optimum insulation thicknesses vary between 20 and 170 mm, energy savings between 22% and 79%, and payback periods between 1.3 and 4.5 years. In addition, he observed variations in the results for different cities in the same climate zone which led him to conclude that the optimum insulation calculations should be done separately for each city and not for a whole climate zone.

Further step in determining optimum insulation thickness is to analyse the influence of both heating and cooling loads on energy consumption. Bolattürk (2008) found that optimization of insulation thickness as a function of cooling loads is more appropriate for energy savings in warm zones. Kim and Moon (2009) did a series of parametric simulations to quantify the impact of insulation of various parts of building envelopes on building energy consumption. They performed simulations by using two different climate weather files, cold and hot. One of the conclusions was that there is a point after which further improvement of U-values provides negligible energy savings. In addition, they stated that, in cold climates, insulation is primarily beneficial for reducing heating energy consumption in winter, while there is no practical benefit for saving cooling energy in summer. On the other hand, in hot climate, only minimum insulation is needed because, according to them, any additional insulation has no impact on reducing heating or cooling energy consumption.

A similar but more comprehensive study of evaluating the impact of insulation level on building total energy consumption and on number of hours when indoor temperature causes discomfort was carried out by Chvatal and Corvacho (2009). In addition to insulation, they analysed the coupled influence of other parameters which can reduce overheating such as shading and ventilation strategy. In office buildings, discomfort level rose with increments in insulation, mainly due to high internal gains, although more shading and proper ventilation strategy may reduce it. Besides the discomfort issue, they also reported that the increase of insulation results in higher total energy consumption as energy used for air conditioning overrides the heating savings. The overall recommendation from their study was to avoid using highly insulated buildings if they accommodate office activities with high internal gains. A quite similar conclusion can be found in the Report on carbon reductions in new non-domestic buildings (DCLG, 2007). Lowering U-values resulted in increasing the CO_2 emissions as the improvement in insulation levels kept the heat in, and created higher cooling requirements. Masoso and Grobler (2008) also proved that rules of thumb such as "better insulation will save energy" are not universally applicable. They found in their simulation that when the cooling set point is over a certain value, depending on the average temperature during the cooling season, more insulation causes cooling load to rise rather than decrease.

2.3.1.2. Thermal mass

Thermal mass of a building and ability to absorb and store heat is highly influenced by materials used in a building structure. The use of dense structural elements as part of building fabrics, such as brick, stone and concrete, increases a building thermal mass. Walls, floors and ceilings made of dense materials interact with the internal environment which have several positive effects on the indoor conditions and energy consumption (Braham et al., 2001). During a winter period, on sunny days, solar heat gains are absorbed by thermal mass and later slowly released. There are two advantages from this; overheating during the high solar radiation periods of the day is avoided, released heat during the late afternoon and early evenings reduces the heating load. In summer, heat stored in the thermal mass reduces peak cooling loads which could result in the use of a smaller HVAC plant. The ability to store and shift the loads also increases a thermal comfort by minimizing indoor temperature variations in both summer and winter.

Overall, thermal mass of the building can contribute to energy savings, both heating and cooling. Aste et al. (2009) did parametric analysis on the influence of the external wall thermal inertia on the energy performance of well insulated buildings. They varied construction elements in the external wall by keeping the same U-value and found that up to 10% reduction in heating energy consumption can be achieved in buildings with a high thermal mass. The impact on cooling energy consumption was even larger; the reduction reached about 20%. The 20% saving in the energy consumption for cooling was also reported by Balaras (1996) who did a review of a lot of studies which investigated the relationship between thermal mass and indoor air temperature, and the effect of thermal mass and night ventilation on cooling demand. In addition, he concluded that even higher reduction in the energy consumption of HVAC systems could be achieved by using night time ventilation or pre-cooling during off-peak hours, in particular in buildings which are unoccupied during the night, such as office buildings and schools. An overview of research studies related to the night time ventilation and the mechanical pre-cooling was provided by Braun (2003). He reported that there is a significant savings potential for use of building thermal mass. However, the savings potential is very sensitive to many factors such as utility rates, type of equipment, occupancy schedule, building construction, climate conditions, and control strategy.

Degree of the positive effect of thermal mass is highly influenced by a location of insulation layer in an external wall and insulation thickness (Asan, 1998). Kossecka and Kosny (2002) analysed the effect of insulation location on heating and cooling load in a continuously operated building with a high thermal mass. Their study showed that differences in total energy demand between least-efficient wall configuration, which is a wall with insulation placed close to the internal surface, and the most effective configuration with externally placed insulation layer might exceed 11% depending on climate. Additional improvement can be obtained by replacing single layer of insulation with two layers which are placed separately. According to Asan (2000), close to optimum solution is to place half of the insulation in the mid-centre plane of the wall and the another half in the outer surface of the wall. He also explicitly claimed that the insulation as a whole should never be used in any other location than external surface of the wall. Similar conclusions was reported by Ozel and Pihtili (2007) who even further extended analysis by splitting insulation into three layers. In that case, the best result was achieved by placing three equal thickness insulation layers on the indoor surface, on the outdoor surface, and in the middle of wall.

2.3.1.3. Glazing

The parameter which also has a significant impact on building thermal behaviour is the size and orientation of glazing. Increased glazing area results in higher gain from solar radiation, which can be beneficial during a heating period but during a summer period affects cooling demand and can lead to overheating. In addition, due to difference in U-values between glazing and external wall to the detriment of the glazing, conductivity gains/losses of the whole envelope are also increased. On the other hand, perimeter areas benefit from the higher percentage of glazing in terms of daylight.

However, large amount of daylight, which enters a space through highly glazed areas, often reduce the quality of visual comfort due to glare problems.

All these issues have been studied in detail and results have been widely published. Bokel (2007) studied the effect of window position and size on the energy demand for heating, cooling and electricity and concluded that an optimal window size is around 30% of the façade area and preferable position of the window should be a top half of the façade. He also reported that a trend in cooling demand follows the increment in window size, while in the case of heating demand there is an optimum minimum around 50% of window size. Lighting load is highly affected by window size for glazing areas up to 50% while for the larger window sizes the advantage of larger area is negligible. Poirazis et al. (2008) investigated the impact of glazing area on the energy consumption of office building with 30%, 60% and 100% window to external wall area. Their study showed that the total end use energy consumption increased by 23% for the 60% glazed building and 47% for the 100% glazed building in comparison with energy end use of 30% glazed building. These results were obtained by using clear glazing in all three cases. The difference in total energy end use can be reduced by implementing glazing units with lower thermal transmittance and total solar transmittance. This measure resulted in „only" 15% higher total energy end use of 100% glazed buildings when compared to the 30% glazed building with clear glazing.

Influence of orientation is significant in the case when facades are different in terms of window-to-wall ratio, or building has one dominant façade. Tzempelikos and Athienitis (2005) investigated the effect of window size on daylighting, peak heating/cooling loads, and overall energy consumption. They used a single perimeter office space with one exterior wall as a base case and came to conclusion that the effect of orientation on heating load is small. The difference between a south façade and north façade did not exceed 13%, which was explained as a result of small solar effects for heating design day. However, the influence on cooling load is significant and the difference between east/west and south facing façade is approximately 17%, while in the case of north orientation the cooling load for a south-facing façade is two or three times higher. On the other hand if facades are identical, the orientation does not have much effect on the energy use on a building level (Poirazis et al., 2008). Aksoy and

Inalli (2006) rotated variously shaped test models from 0° to 90° with 10° step between each position and found that the difference between the minimum and maximum yearly heating energy consumption is about 5%.

2.3.1.4. Solar gains control measures

Since the solar heat gains and the internal heat gains are major components of office building cooling loads, a number of measures to reduce these heat gain sources can be implemented.

One way of reducing the solar heat gains is by using different types of shading systems which can be installed externally or internally. External shading elements can be horizontal, vertical, or combination of both. In addition, shading elements can be movable or fixed. The effectiveness of this type of shading depends on the type of shading and its placement relative to openings. Internal shading devices, such as Venetian blinds or curtains, have limited ability to control solar gain and usually are less effective than external shading devices. However, these interior devices can contribute to visual comfort in the work place through glare control. Some shading systems, such as blinds, can be located between glass panels in double or triple glazed windows.

Overhangs, if precisely sized, can lead to significant cooling energy reductions, while having negligible effect on reduction of beneficial solar heat gains received during winter period. Raeissi and Taheri (1998) presented a model which determines the effect of window overhangs on building energy consumption. For optimum overhang dimensions they calculated a 12.6% cooling load reduction, while heating loads were increased by only 0.6%. The outputs were validated by measured data. Presented results were given for specific climate and latitude. However, the effect of overhangs on overall cooling/heating energy consumption varies with different climate conditions and geographical position as well as window characteristics such as size and properties of glazing. Manzan and Pinto (2009) did an optimization study in which an external shading device in office building were optimized with main objective to minimize primary energy consumption for heating, cooling and lighting. Four external shading variables (height above window, depth, angle and distance from wall) were calculated for several glazing systems (standard glass, high performance glass, window with/without reveals). Each configuration required different optimum external shading geometry providing savings in primary energy consumption between 5% and 17%.

Another way of reducing solar heat gains is by careful design and selection of glazing elements with lower values of solar heat gain coefficient (SHGC). This can be achieved by using glazing either with high reflective capabilities or with high heat absorption characteristics. Cordoba et al. (1998) did a comparison of the influence of various glazing types on the office building energy consumption in Madrid. Reference glazing which was used in comparison was composed of clear inner glass and outer heat absorbing tinted glass with total solar heat gain coefficient of 0.48. By replacing the reference glazing with reflective outer glass and solar heat gain coefficient of 0.26, they calculated 10% reductions of the annual energy consumption. In similar study, Stegou-Sagia et al. (2007) compared the energy consumption of a 20% glazed office building with clear glazing units with the same building equipped with grey tinted glazing units. Reported reductions in total energy consumption were 10% and 6.5% for Athens and Thessalonica respectively.

2.3.1.5. Infiltration

Infiltration is the uncontrolled flow of outdoor air into a building through cracks and other unintentional openings and mainly depends on wind direction and pressure, temperature differences between outdoor and indoor air, construction type and quality, and occupant use of exterior doors and operable windows. Infiltration is also known as air leakage into a building or building air tightness. Uncontrolled air infiltration causes heat loss in winter and heat gain in summer, which results in higher energy consumption. Emmerich and Persily (1998) studied the energy use in commercial buildings due to infiltration. The energy impact of infiltration in U.S. office buildings was estimated based on the analysis of a set of 25 buildings developed to represent the U.S office building stock. They showed that infiltration is responsible for about 13% of the heating load and 3% of the cooling load. In newer buildings, due to the higher level of insulation, the impact of infiltration is even higher and accounts for about 25% of the heating load and 4% of the cooling load. The overall conclusions of this study was that the total annual energy impact of infiltration in U.S. office buildings is 15% of the total heating energy and 4% of the total cooling energy.

The airtightness of a building envelope can be given as either a measure of air flow rate through the building envelope in $m^3/s{\cdot}m^2$ at a specified pressure difference or as a predicted overall rates of infiltration using the number of Air Changes per Hour (ACH). Measured air flow rate is usually given at indoor-outdoor pressure difference of 50 Pa in the United Kingdom, or at 75 Pa in the U.S. There is limited data available on real airtightness levels in buildings. Persily (1998), based on a review of published literature, assembled and evaluated measured envelope airtightness data for 139 buildings. The reported mean airtightness value for all buildings was 27.1 $m^3/h{\cdot}m^2$ at 75 Pa, but the range and standard deviation were large. Emmerich and Persily (2005) updated the previous analysis by including data from over 100 additional buildings. The overall average airtightness of 28.4 $m^3/h{\cdot}m^2$ at 75 Pa was very close to the value reported by Persily in 1998. In the most recent study, Emmerich et al. (2007) divided the data into north and south subsets for the North American buildings only. They also excluded buildings older than 1960, industrial buildings, and extremely leaky buildings which led them to an average airtightness at 75 Pa of 23.76 $m^3/h{\cdot}m^2$ for north subset and 42.48 $m^3/h{\cdot}m^2$ for south subset. VanBronkhorst et al. (1995) summarized infiltration rates for 25 buildings which represent the U.S office building stock. Infiltration rates were generated for a wind speed of 4.5 m/s based on the age and height of the building and the average annual temperature difference. The minimum reported infiltration was 0.16 ACH, while the leakiest building had infiltration rate of 1.0 ACH.

Sharples et al. (2005) tested for airtightness a warehouse in the UK with nearly 58,000 m^2 floor area. The measured air permeability was 2.25 $m^3/h{\cdot}m^2$, substantially lower than required by the regulations. They also compared results with similar studies which measured air permeability of very large buildings and concluded that the target value of 10 $m^3/h{\cdot}m^2$ specified by current UK building regulations is readily achievable by using standard techniques.

2.3.2. Environment and activity related parameters

The main purpose of a building and its HVAC system is to provide acceptable indoor conditions for the wellbeing of occupants. Inadequate indoor conditions affect health, productivity and comfort of the occupants. In office building environments, poor indoor conditions lead to decreased productivity and increase illness-caused absence. This could be avoided by adequate spending on improving and maintaining the most satisfactory indoor environment. Seppänen and Fisk (2006) reviewed the literature on the effect of indoor environment on health and productivity and found that there is a strong relationship between: ventilation rates and short-term sick leave, ventilation rates and work productivity, perceived air quality and productivity, temperature and productivity, and temperature and sick building syndrome symptoms. It is obvious that special attention has to be paid to an indoor thermal comfort and an indoor air quality. That is the reason why the minimum values, which satisfy indoor thermal comfort and air quality, are defined by various national and international standards.

Beside indoor conditions, internal heat gains are another important parameter which has significant impact on energy consumption both directly, through lighting and equipment loads, and indirectly by affecting heating and cooling loads. They can be the dominant reason why in the temperate climates, such as in the UK, cooling systems in office buildings exist (Jenkins, 2009). The main sources of internal heat gains are:

- Occupants,
- Office electrical equipment, and
- Artificial lighting.

Benchmark values for internal heat gains are mainly based on measured data collected in numerous surveys of different building types and activities. When measured data is not available, the common way of obtaining values for internal heat gains is to use empirical values, based on experience, which are considered good practice in the industry (CIBSE, 2006).

Internal heat gains, in particular from artificial lighting can be reduced by implementing daylight control. Many results related to energy savings due to daylight control has been presented in literature. Lam and Li (1998) proposed a simple method for estimating energy savings of electric lighting and cooling. In their case study, which was based on generic office building in Hong Kong, daylight could maintain sufficient level of indoor illuminance for about 40% to 60% of the time, which resulted in 50% reduction in artificial lights electricity consumption and additional 11% electricity savings for cooling by assuming the coefficient of performance (COP) of 3 for the chiller plant. Same authors did field measurements for several fully air-conditioned cellular offices facing opposite orientation with and without daylight control (Li and Lam, 2001). Measurements confirmed that energy savings in electricity for artificial lighting could be up to 50% for the perimeter offices. Li et al. (2006), in the similar study, found that the percentage of saving is slightly lower for an open plane office spaces and amounts to around 33%.

Roisin et al. (2008) evaluated the performance of different daylight control systems for three locations in Europe and the four main orientations. The parametric study showed that the potential saving could vary from 45% to 61%. The former value was obtained for the north-facing office room in Stockholm, while latter was calculated for the south-oriented office room in Athens. Bodart and De Herde (2002) analysed the impact of daylight energy savings on the total energy consumption in office buildings by varying façade configuration, orientation and internal walls reflection coefficients. They found that daylight control could reduce the total primary energy consumption (energy for heating, cooling, humidification, dehumidification and artificial lighting) for around 40% for a typical office building and up to 50% for a building with high performance glazing. At the same time, the artificial lighting electricity savings were between 50% and 80%. Similar level of reductions was reported by Knight (1999) who measured between 44% and 76% of daylight-linked savings depending on the type of a daylight control. Lee and Selkowitz (2006) also reported significant savings in artificial lighting electricity consumption, up to 60%, based on nine-month monitored field study of the New York Times Headquarters daylighting mockup.

2.4. Office building models in this research

Following on from the outputs of the “four towns” survey and proposed non-domestic built form classification and the literature review, the models of four office building types were developed to represent the most typical office building built forms. The first office building type represents open-plan sidelit buildings (OD). The second building type is the cellular sidelit building (CS). The third type is artificially lit open-plan building (OA) and the last type symbolised the composite sidelit cellular around artificially lit open-plan (CDO) built form category.

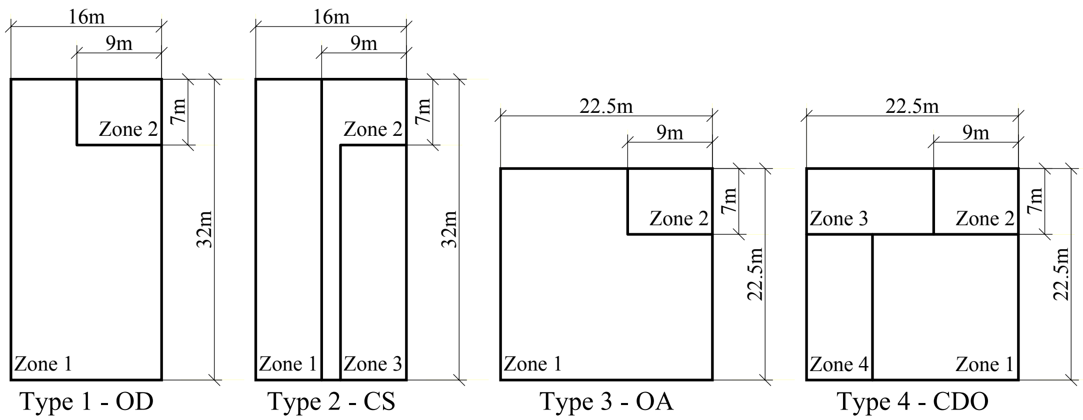


Figure 2.8. Office building model types

Each storey in office building models is composed of office areas and common spaces. Common spaces (Zone 2 in diagrams in Figure 2.8) represent areas such as reception areas, toilets, tea kitchens, circulation space, etc. Diagrams of the four building types can be seen in Figure 2.8. Floor area of the each storey is roughly the same in all models and amounts around 510 square meters. Building types one and two are narrow plan buildings with a 32 by 16 meters footprint. Both buildings are sidelit and differ only in the office space arrangement. The office space in the building type one consists of one large open space (Zone 1), while the office space in the building type two is divided by corridor into two zones of cellular offices (Zone 1 and Zone 3). Building models three and four are square buildings with a 22.5m by 22.5m footprint. In the building model three, the floor plane is dominated by a single large open-plan office area (Zone 1). The last model, building type four, has each storey divided into four zones. The main zone, Zone 1, has an open-plan office arrangement, while Zones 3 and 4 represents cellular office layout. Zone 2, as in the previous building types, is reserved for common areas.

The floor layouts presented in Figure 2.8 represent the office building models thermal zones used in both building and HVAC systems cooling/heating demand calculations. Thermal zoning is important, since it has impact on demand calculations as well as on the performance of multi-zone HVAC systems and required HVAC equipment capacity (ASHRAE, 2009). There is a need for zoning when rooms or areas within a building have differences such as heating/cooling temperature setpoints, activity levels, occupancy pattern, etc. This was the reason why each floor in the developed office building models was divided into office zones and a common area zone. Also, where several rooms or areas of a building behave in a similar manner, they can be grouped together as a „zone", which was exactly the case in creating cellular office zones in abovementioned building models. On the other hand, large open-plan spaces can be divided into perimeter and internal zones, despite there are being no physical partitions, as internal zones are not affected by the weather (ASHRAE, 2008, Stocki et al., 2007). The depth of the perimeter zones is mainly affected by the exposure to the sunlight and daylight which depends on zone orientation, shading, window size, characteristics of the glass, etc. CIBSE Guide B (CIBSE, 2005a) suggested that most deep plan buildings should be divided into perimeter and internal zones. However, the Zone 1 in building models 1 and 3 is not divided in perimeter and internal zones since the depth of building models is not large.

All four building types are three storeys high with floor-to-ceiling height of 3.5 meters. The main reason of having three storeys is the possible unequal energy demand each storey can have due to its interactions with the environment. First storey energy demand is affected by the heat exchange with the ground through the floor. On the other hand, the top storey is exposed to additional solar heat gains through a roof element. Moreover, a transmission heat transfer area is increased by the area of the roof. The intermediate storey usually does not interact with the external environment through floor and ceiling elements, which means that if a floor layout is unique for a whole building and if conditions above and below are identical these two elements can be assumed to be adiabatic surfaces. Energy consumption of intermediate storeys in buildings with more than three storeys can be determined by calculating energy consumption of one intermediate storey only. Total consumption can be determined by multiplying this by the number of intermediate storeys.

One of the main tasks in this research it to analyse the influence of various parameters, individually or in combination, on a building thermal behaviour. For the purpose of this research, some parameters were varied and some of them were kept constant.

Building envelope parameters such as construction U values, percentage of glazing and orientation were varied, while building thermal mass, position of insulation and infiltration rate were fixed. Indoor thermal conditions and air quality parameters were chosen to comply with various national and international standards. Values for internal heat gains from occupants, office electrical equipment and artificial lighting were based mainly on good practice guides and benchmark values.

Daylight and solar control measures were applied to base models in order to evaluate their impact on energy consumptions. Included measures were categorised into two groups: solar heat gain control measures and internal heat gain control measures. The former category included overhangs attached above glazing elements and typical glazing replaced with more advanced glazing with reflective characteristics, while the latter category was based on daylight control. To include the possible impact a building orientation could have on its thermal behaviour, models were rotated at 45-degree intervals.

The following sections give details of various office building model parameter selection criteria.

2.4.1. Building envelope parameters

Nowadays, the quality of both new-build and refurbished buildings is ensured by achieving a minimum standard for building work which is specified by national building regulations. The regulations set standards for the design and constructions of buildings with the main purpose of ensuring health, safety, welfare and convenience of people in and around buildings. They are also designed to further the conservation of fuel and energy. In general, building regulations are related to the technical aspects of construction such as structural stability, fire resistance, escape routes, disabled access, weather resistance, thermal insulation, and drainage.

2.4.1.1. Historical review of the UK building regulations

The first building construction legislation within England was introduced in 1667, shortly after the Great Fire of London which happened in 1666. It emphasized the need for taking into consideration possible spread of fire between buildings during the rebuilding work since the rapid spread of the fire through timber buildings built next to each other contributed greatly to the devastation caused by the Great Fire of London. The introduced legislation, the London Building Act, required buildings to achieve some degree of fire resistance. During the period of the industrial Revolution in the 18th and 19th century towns were expanding very quickly creating poor living and working conditions in densely populated urban areas. Inadequate sanitation, damp conditions and lack of ventilation were some of the reasons which led to outbreak of Cholera and other serious diseases. The Government reaction to this was the introduction of the first Public Health Act in 1875, The Act required from urban authorities to make bylaws for new constructions with special attention to:

- the structure of buildings, ensuring stability and prevention of fires
- the drainage and provision of air space around buildings, to ensure health considerations

However, as the Public Health Act was simply a guide for local authorities to make their own bylaws, the standards across the country varied greatly. The Act had two major revisions in 1936 and 1961, which led to the introduction of the first set of national building standards in England and Wales, the Building Regulations 1965.

The Building Regulations 1965 (Gov.Legislation, 1965) came into operation in February 1966 throughout England and Wales, with the exception of the Inner London Boroughs where the London Building Acts continued to prevail. Part F of the Regulations introduced the thermal insulation requirements where the maximum values of the overall heat transfer coefficient (U-value) for major construction elements were specified. Roofs and floors were limited to have the U-value not more than 1.42 $W/m^2{\cdot}K$ (0.25 $BTU/h{\cdot}ft^2{\cdot}F$ in the original document), while for external walls this value was limited to 1.70 $W/m^2{\cdot}K$ (0.30 $BTU/h{\cdot}ft^2{\cdot}F$). In addition to the Part F, the Schedule 11 listed several tables which specified the type of constructions, in particular type of roof, type of floor and type of insulation, as same as the construction of external wall, which would satisfy the requirements of regulations. The next regulations, The Building Regulations 1972 (Gov.Legislation, 1972), brought several changes but none of them was related to the thermal insulation which means that the U-value limitations remained the same as in the 1965 Regulations.

The Building Regulations 1976 (Gov.Legislation, 1976) came into power in January 1977 and was also applied through England and Wales apart from the Inner London Boroughs. It extended the part related to the thermal insulation. Maximum U-values of different building elements were significantly improved in comparison with the previous Building Regulations. Roof, including any ceiling to the roof or any roof space and any ceiling below that space, had to have the maximum U-value of 0.6 $W/m^2{\cdot}K$ while both floors and external walls had U-value limited to 1.0 $W/m^2{\cdot}K$. Furthermore, the calculated average U-value of perimeter walling was set not to exceed 1.8 $W/m^2{\cdot}K$. For calculating a perimeter walling average U-value, the Regulations recommended the U-value of any window opening to be assumed 5.7 $W/m^2{\cdot}K$ for single-glazed or 2.8 $W/m^2{\cdot}K$ for double-glazed.

The Building Regulations 1976 were replaced by the Building Regulations 1985 (DEWO, 1985) which Part L covers the topic of conservation of fuel and power. For the first time in the Regulations, buildings were divided into dwellings and buildings other than dwellings which resulted in different requirements for each of these two categories. The maximum U-values for office building were set to 0.6 $W/m^2{\cdot}K$ for exposed walls, roofs and exposed floors, and 5.7 $W/m^2{\cdot}K$ for windows and rooflights. In addition to construction elements U-values, the maximum single glazed area for office buildings was limited to 35% of exposed wall area. Double glazed areas were permitted to have up to twice the single glazed areas, while glazed areas could be even more increased in case of using a double glazing with low emissivity coating or triple glazing. Suggested U-value of windows and rooflights to be used in calculation was 5.7 $W/m^2{\cdot}K$ if single glazed, 2.8 $W/m^2{\cdot}K$ if double glazed and 2.0 $W/m^2{\cdot}K$ if triple glazed or double glazed with low emissivity coating. The Part L 1990 edition of the Building Regulations 1985 (DEWO, 1990) brought significant improvement in insulation of the building fabric. U-values of exposed walls, exposed floors, ground floors and roofs were not allowed to exceed 0.45 W/m^2·K. However, single glazed windows and rooflights were still allowed. Requirements regarding the maximum glazed areas remained the same as in the previous Building Regulations.

The Approved Document L 1995 edition of the Building Regulations 1991 (DEWO, 1995) replaced the 1990 edition. The standards of fabric insulation were improved by changing the method of calculating U-values, which took into account thermal bridges such as mortar joints, timber joints and studs, despite the actual values of U-values for exposed walls, exposed floors and ground floors remained the same as in previous standard; 0.45 W/m^2·K. The roof maximum allowable U-value were improved to 0.25 W/m^2·K, although U-value of 0.45 W/m^2·K was acceptable for a flat roof or insulated sloping roof with no loft in buildings other than dwellings. U-value of 3.3 W/m^2·K became a standard for windows, doors and rooflights which were achievable by installing windows with sealed double-glazed units or by implementing secondary glazing. The window area allowances were changed, and for office buildings it amounted to 40% of exposed wall area for windows and doors and up to 20% area for rooflights. Increases over the basic area allowances were also permitted, but only if compensating provisions were made. This standard, when compared to previous regulations, expanded requirements by introducing two new measures: limiting infiltration, and lighting efficiency. Infiltration of cold outside air through leakage paths in the building envelope was recognized as a factor which significantly affects space heating demand. Due to that, several sealing measures which implementation can limit leakage were listed in the standard. Artificial lighting systems were covered in a new section in which guidance on achieving the requirements for lighting efficiency and controllability was given. The Standard proposed several types of high efficiency lamps to be used and defined the practical aim of lighting control which should be implemented to achieve the maximum use of daylight and to avoid unnecessary lighting during the unoccupied hours.

The Approved Document L2 2002 edition of the Building Regulations 2000 (DTLR, 2002b) dealt with the buildings and parts of buildings other than dwellings. This was for the first time that building regulations addressed separately dwellings and buildings other than dwellings in different documents. Dwellings were regulated by

Approved Document L1 2002 edition (DTLR, 2002a). Standard U-values for non-domestic buildings were improved in comparison with the previous Part L 1995. The maximum U-value of walls, including basement walls, was set to 0.35 $W/m^2 \cdot K$, while the U-values of floors, including ground floors and basement floors, and flat roofs were limited to 0.25 $W/m^2 \cdot K$. Furthermore, the U-values of windows and rooflights were significantly refined and were not allowed to exceed 2.0 $W/m^2 \cdot K$ for glazing in wood or PVC frames or 2.2 $W/m^2 \cdot K$ for glazing in metal frames. Maximum area of openings remained the same and amounted to 40 per cent for office buildings. However, in order to provide greater design flexibility, the U-values of construction elements and the areas of openings were allowed to vary from the standard U-values if suitable compensating measures were taken, although the poorest U-values acceptable when trading off between construction elements were set to 0.35 $W/m^2 \cdot K$ for parts of roof and 0.7 $W/m^2 \cdot K$ for parts of exposed wall and floor. To decrease the influence of air infiltration on building energy consumption, the building air leakage standards were further expanded. The design air permeability was not allowed to exceed 10 $m^3/h \cdot m^2$ at a pressure difference of 50 Pa. One important novelty in the Part L2 2002 was the set of recommendations which should help avoiding solar overheating.

In the meantime, the European Parliament accepted Directive 2002/91/EC (EC, 2003) on the Energy Performance of Buildings (EPBD) which requires Member States to take the necessary measures at greenhouse gas emissions reduction and compliance in energy requirements. The main requirements of the EPBD are to develop and implement energy calculation methods based on an overall energy performance, and to define minimum energy requirements for both new and refurbished buildings. In addition, the energy certificate for new and existing buildings, obtained from independent experts, is mandatory when they are constructed, sold or leased. The certificate must include details about current building energy efficiency as well as the recommendations about possible improvements in energy performance. Another important requirement of the EPBD is that inspections of HVAC equipment became compulsory.

The EPBD requirements were included in the approved document L2 2006 (ODPM, 2006b) which was the successor of the Part L2 2002. Part L2 2006 came into effect in April 2006 and was in power until October 2010 when it was replaced by the approved document L2 2010 (ODPM, 2010). However, the requirements regarding standard U-values and design air permeability in both regulations remained the same as in the Part L2 2002. Table 2.2 summarised changes in maximum allowed building elements U-values in the UK Building Regulations starting with the first Regulations from 1965 up to the latest Part L2 2010 standard.

Table 2.2. Historical review of maximum allowed building elements U-values in Building Regulations

Building Element	U-value [W/m^2·K]						
	1965/72	1976	1985	1990	1995	2002	2006/10
External Wall	1.7	1.0	0.6	0.45	0.45	0.35	0.35
Roof	1.42	0.6	0.6	0.45	0.25/0.45	0.25	0.25
Ground Floor	1.42	1.0	0.6	0.45	0.45	0.25	0.25
Glazing	n/a	5.7	5.7	5.7	3.3	2.0/2.2	2.2

2.4.1.2. Building models U-values

In order to represent buildings constructed through different periods in the past, and at the same time to analyse the influence of U-values on building energy consumption, five types of building fabrics were prepared and attached to building models. First building fabric type (BF1) has no insulation at all. Beside the absence of insulation, buildings in BF1 category are equipped with single glazing windows. Second building fabric type (BF2) has low level of insulation and slightly better glazing than BF1. In general, models with building fabrics two represent pre-1990 buildings. Next building fabric type (BF3) complies with both Part L 1990 and Part L 1995, and with its medium level insulated envelope characteristics covers buildings constructed between years 1990 and 2002. Fourth building fabric type (BF4) includes buildings constructed in accordance with Part L 2002 building regulations and have high level of insulation. The last building fabric type (BF5) can be characterized as a current best practice as U-values of building fabric components are significantly lower than required by current UK building regulations. The U-values of major building elements, which are external wall, flat roof, ground floor and glazing, of each of these five building fabric types can be seen in Table 2.3. These values were calculated by EnergyPlus building simulation software by taking into account physical properties of building fabric elements. More about building fabric characteristics, such as construction element layers and thermo-physical characteristics of materials used in building elements, can be found in Appendix A.

Thermal mass of office buildings in this research can be classified as heavy according to the classification provided by Rennie and Parand (1998) where the building thermal mass is classified into four categories: very light, light, heavy and very heavy. In addition, office building models were made with a single insulation layer. Insulation layer in external walls and roof, where it is present, was positioned close to the external surface, as this measure was proved to have positive effect on energy savings, especially in combination with high thermal mass.

Table 2.3. Building Fabrics (BF) U-values

Building Element	U-value [W/m^2-K]				
	BF1	BF2	BF3	BF4	BF5
External Wall	1.62	0.54	0.40	0.32	0.24
Flat Roof	2.48	0.43	0.31	0.17	0.14
Ground Floor	1.03	0.82	0.34	0.25	0.14
Glazing	5.87	3.15	2.73	1.92	1.78

2.4.1.3. Building models glazing ratio

The percentage of glazing has been varied during the architectural history on non-domestic buildings. In the first quarter of the 20th century, typical building glazing ratio was below 20% (Gakovic, 2000). One of the key reasons for a low level of glazing was poor features of the glass used in glazing units. Namely, fenestration areas were the main heat sinks and designers tried to minimize it to keep heat losses as low as possible. Also, the glass was quite an expensive material. By improving the quality of glazing and reducing the cost, the percentage of glazing area started to increase. Moreover, the desire to create a bright space with a light and transparent facade has led architects to design highly glazed buildings, which makes their energy efficiency questionable. All of this created large non-uniformity in non-domestic building stock concerning the percentage of glazing.

Gakovic (2000), in his study of areas and types of glazing and other openings in the non-domestic building stock, reported correlations between area of openings and the building external wall area and floor area. Findings were based on openings measurement and classification done during field survey. Buildings which were analysed were a subset of the survey on energy use in non-domestic buildings done by a team from Sheffield Hallam University (Mortimer et al., 2000), which were a subset of the "four towns" survey (Brown et al., 2000). Overall, data which were analysed were collected from 101 locations with the total floor area of just above 167,000 m^2. Close to 46% of all area belonged to the cellular sidelit building type (Type 2 – CS). The second largest category was composite sidelit cellular around artificially lit open-plan building type (Type 4 – CDO) with 34% of the total floor area. Open-plan sidelit building type (Type 1 – OD) with slightly above 6% and artificially lit open-plan building type (Type 3 – OA) with 1.6% of the total floor area accounted significantly lower than previous two categories.

The selected 101 locations were further broken down into three categories: "traditional", "framed", and "shed". "Framed" and "shed" categories were additionally subdivided into smaller groups. The "Framed" category was composed of following subgroups: "framed, curtain wall", "framed, deep plan", and "framed, other". The "sheds" category consisted "sheds with rooflights" and "sheds, other" subgroups. Once analysed, results showed the strong correlation between the gross areas of openings and the total gross floor area for each of these subcategories. The highest ratio of glazing to floor was for "framed, curtain wall" subcategory and amounted to 0.29, while the lowest one, 0.08, was for the "framed, deep plan" buildings. Gakovic also reported the percentage of glazing areas in the external walls. The highest mean percentage of glazing was around 60% for "framed, curtain wall" buildings. This value was between 14% and 25% in most other subgroups.

By taking into account results from the survey and the need to include broad range of different levels of glazing in order to investigate the influence of various levels of glazing on office buildings energy consumption, three different percentages of glazing were coupled to each of building models described previously in chapter 2.4 and defined in Figure 2.8. The 25% of glazing represents buildings with low level of glazing. The 50% of external area covered with glazing is a typical value for medium glazed buildings, while buildings with 75% of glazing can be classified as highly glazed buildings. The quantity of glazing is equally distributed on all facades. Extremes, such as completely glazed buildings or quite new and not very common building envelopes which can be found in double skin façade buildings, were not included in this research study.

2.4.1.4. Solar heat gain control

As already mentioned, solar heat gains are one of major components of the office building cooling load, which can be reduced to some extent by implementing various solar heat gain control measures. Two particular measures were included in this research: typical glazing replaced with reflective glazing and overhangs attached above windows.

Building models in this study were initially developed with regular window type which have clear glass outer pane with high gain coefficient (SHGC) and light transmittance. As these values of SHGC and light transmittance can cause glare and overheating, the building models were also created with advanced reflective glazing. Solar heat gain coefficient and light transmittance factor of a reflective glazing are significantly lower in comparison with regular glazing. However, the overall window U-value is almost unchanged as it can be seen in Table 2.4 in which glazing parameters of two window types were compared.

Table 2.4. Regular/Reflective glazing properties

Glazing type	BF1		BF2		BF3		BF4		BF5	
	Reg.	Ref.	Reg.	Ref.	Reg.	Ref.	Reg.	Ref.	Reg.	Ref.
SHGC	0.847	0.505	0.74	0.435	0.742	0.432	0.631	0.362	0.637	0.365
Light transmittance	0.892	0.335	0.801	0.312	0.801	0.312	0.761	0.296	0.761	0.296
U-value [$W/m^2 \cdot K$]	5.87	5.81	3.15	3.13	2.73	2.71	1.92	1.91	1.78	1.77

The other option of solar gain control is implementation of external shading elements, in particular window overhangs. Selection of the most suitable window overhang requires a comprehensive analysis. Results are mostly case-limited and cannot be easily replicated since depends on many design factors such as latitude, climate, solar radiation transmittance, illuminance levels, window size and type, etc. Due to absence of a generic solution for size and position of window overhangs, and in order to maintain consistency among created office building models, only one type of window overhang was implemented in this research. Horizontal overhangs were placed twenty centimetres above a window with a depth of 0.7 meters. Since windows in models were represented by single strip of glazing per each external wall, the influence of overhang left/right extension from window is negligible, so these values were set to zero. Position and size of the window overhang used in this study can be seen in Figure 2.9.

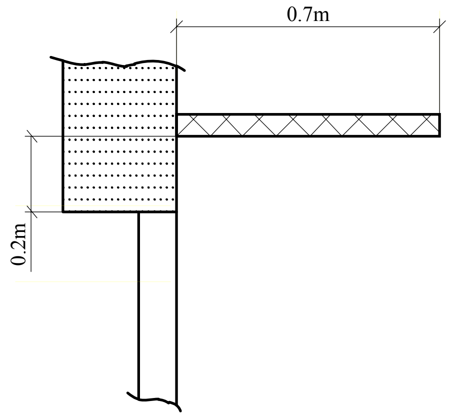


Figure 2.9. Overhang geometry

2.4.1.5. Air infiltration rate

Air infiltration is a very complex topic and, as it can be seen from many research studies, it is very difficult to predict the air change rate which will correctly represent the real situation in building. Selecting the realistic infiltration rate is much easier if there are measured data. However, these data can be obtained once building is built and tested and not really at the design stage. Nowadays, the infiltration rate is limited by the Building Regulations by setting the maximum value of air leakage. The UK Building Regulations Part L2 2002 introduced the requirement for building envelopes to attain a reasonable standard of airtightness for buildings with floor areas more than 1000 m^2. A reasonable standard is defined as a leakage of no more than 10 m^3/h per m^2 of total building envelope surface area for a pressure difference of 50 Pa.

CIBSE Guide A (CIBSE, 2006) subclassified buildings according to the airtightness into five categories:

- 20 $m^3/h{\cdot}m^2$ at 50 Pa – an existing „leaky" building that does not comply with current regulations,
- 10 $m^3/h{\cdot}m^2$ at 50 Pa – a building which complies with current standards,
- 7 $m^3/h{\cdot}m^2$ at 50 Pa – a moderately tight building,
- 5 $m^3/h{\cdot}m^2$ at 50 Pa – a tight building, and
- 3 $m^3/h{\cdot}m^2$ at 50 Pa – a very tight building.

For these five categories, approximate estimates of annual average infiltration rate for non-domestic buildings were provided. Average infiltration rates are based on CIBSE hourly 20-year average wind and temperature data and they should be used for heat loss calculation, estimation of contribution of infiltration air to ventilation needs, and estimation of contribution of infiltration air to summer cooling potential or infiltration heat load. Infiltration rate for a leaky air conditioned office building ranges between 0.45 and 0.60 ACH, while for the building which complies with current standard it ranges between 0.25 and 0.30 ACH. A very tight building has an average infiltration rate as low as 0.10 ACH. However, these values are given for normally exposed site and should be increased by 50% on severely exposed sites, or reduced by one third on sheltered sites.

It is most likely that new buildings may be tighter and it is not unrealistic to expect that current best practice building can have infiltration rate as low as 0.1 ACH. On the other hand, single glazed old buildings probably have several times higher infiltration rates. However, the infiltration rate used in this research was kept constant. The selected value of 0.3 air changes per hour complies with the current Building Regulations but also represents a crude approximation due to vast differences in reported measurements, where the infiltration rate varied by factor of ten.

2.4.2. Indoor thermal condition

Careful selection of indoor thermal condition and air quality parameters is crucial so the adequate, desired indoor environment conditions can be maintained. Indoor thermal condition should be carefully designed and controlled in order to provide and maintain occupants" thermal comfort. ASHRAE Standard 55-2004 (ANSI/ASHRAE, 2004) defined thermal comfort as *"that condition of mind which* *expresses satisfaction with the thermal environment"*. There are six parameters which influence an occupant"s thermal comfort:

- Air temperature,
- Mean radiant temperature,
- Relative air speed,
- Air humidity
- Metabolic heat production, and
- Clothing.

The first four are physical parameters, while the latter two are individual personal factors.

Temperature is usually the most important parameter affecting thermal comfort. The room air temperature and mean radiant temperature can be combined as the operative temperature. The operative temperature in environments with low or no air movement, which is typical for office environments, can be calculated by equation 2.1:

$$\boldsymbol{t_o = \frac{t_{a,i} + t_r}{2}} \quad \textbf{2.1}$$

where: t_o is the operative temperature (°C), $t_{a,i}$ is the surrounding air dry-bulb temperature (°C), and t_r is the mean radiant temperature (°C).

The EU Standard EN 15251:2007 (CEN, 2007b) recommends design values of the operative temperature to be used in designing buildings and HVAC systems. The Standard categorises buildings into four categories according to the level of expectation of thermal comfort based on PPD-PMV index (predicted percentage of dissatisfied people at predicted mean vote). Recommended PPD and PMV ranges are given in Table 2.5.

Table 2.5. Building categories according to the level of thermal comfort expectation

Category	PPD [%]	PMV
I	< 6	-0.2 < PMV < 0.2
II	< 10	-0.5 < PMV < 0.5
III	< 15	-0.7 < PMV < 0.7
IV	> 15	PMV < -0.7; or PMV > 0.7

For the normal level of expectation (category II), which should be used for new buildings and refurbishments, the minimum operative temperature during heating period should be 20°C for buildings which accommodate office activities. On the other hand, the operative temperature should not exceed 26°C during cooling operation time. In addition to maximum and minimum design temperatures, indoor temperature ranges for hourly calculation of cooling and heating energy are also recommended. For the category II in buildings with primarily sedentary activity, the temperature range for heating should be between 20°C and 24°C while the recommended temperature range for cooling is between 23°C and 26°C.

Environmental design chapter of CIBSE Guide A (CIBSE, 2006) gives general guidance and recommendation on suitable winter and summer temperature ranges for different building types. Office buildings, according to them, should have winter operative temperature range between 21°C and 23°C and summer operative temperature range between 22°C and 24°C. These values are quite similar to the indoor design conditions recommended by ASHRAE in the HVAC Applications Handbook (ASHRAE, 2007). The only difference is in the summer operative temperature range, as ASHRAE recommends this range to be between 23°C and 26°C.

The operative temperature can be replaced in most cases by the air temperature as a design temperature, unless temperatures of room surfaces differ significantly (CEN, 2007b). CIBSE Guide B (CIBSE, 2005a) drew the similar conclusion that the difference between air and mean radiant temperature for buildings with moderate to good insulation is insignificant and can be neglected. ASHRAE Standard 55-2004 presented in the appendix several conditions, which if they exist, the air temperature can be assumed equal to the operative temperature. The approximation is acceptable if there is no radiant panel heating/cooling system, no heat generating equipment in the space, the SHGC is limited and the average U-value of perimeter walling is determined by equation 2.2.

$$U_W < \frac{50}{t_{d,i} - t_{d,e}} \qquad \textbf{2.2}$$

where: U_W is the average U-value of perimeter walling (W/m^2·K), $t_{d,i}$ is the indoor design temperature (°C), and $t_{d,e}$ is the exterior design temperature (°C).

If the weather data for London is used, equation 2.2 suggests that the maximum U-value of perimeter walling should not be over 1.85 W/m^2·K if the air temperature is to be used instead of the operative temperature.

By taking into consideration previous approximation, the control of indoor environment in this research was based on dry-bulb air temperature by using the dual setpoint thermostat. Following the recommended design temperatures by various standards and institutions, setpoint in office spaces was set to 22°C during heating period and 24°C during cooling period. Common areas were allowed to have lower temperature during heating (20°C) and higher temperature during cooling (26°C). These setpoints are supposed to be met by HVAC system during occupied period, which in office buildings is clearly defined, as buildings are occupied during weekdays between 7am and 7pm only. During unoccupied period, some buildings require a low level of heating to avoid condensation/frost damage or to prevent the building becoming too cold. CIBSE Guide A suggested a temperature of 10°C to be maintained as a general minimum. AHSRAE Standard 90.1-2007 (ANSI/ASHRAE, 2007b) is even more strict and limits setback temperature to 12.8°C (55°F) to be used at night-time, weekends and other holidays during the heating season. Cooling setback temperature setpoint, according to ASHRAE, should maintain zone temperatures below 32.2°C (90°F) in order to prevent the building becoming too hot and to reduce the start-up cooling load the next morning. In this research, during unoccupied hours, thermostat calls for heating if temperature drops below 12°C in any of the zones, while overheating is prevented by turning the cooling on if temperature exceeds 28°C in offices or 30°C in common areas. Figure 2.10 presents the thermostat control operation for a weekday with appropriate setpoint values during occupied/unoccupied period for both offices and common areas.

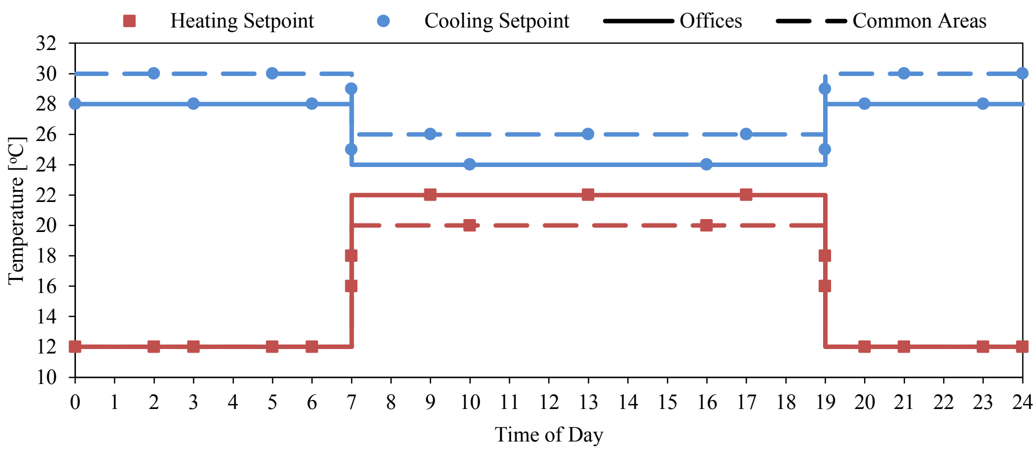


Figure 2.10. Dual thermostat setpoints (offices/common areas)

2.4.3. Indoor air quality

Acceptable indoor air quality is defined by ASHRAE Standard 62.1-2007 as *"air in which there are no known contaminants at harmful concentrations as determined by cognizant authorities and with which a substantial majority (80% or more) of the people exposed do not express dissatisfaction"* (ANSI/ASHRAE, 2007a). The indoor air quality can be achieved by recommended level of ventilation. The required ventilation is based on health and comfort criteria, where in most cases the health criteria will be met by the required ventilation for comfort.

Achieving and maintaining good indoor air quality is an important issue. Office indoor air quality is particularly important as poor air quality can result in loss of productivity, absence from work and in some cases medical problems (Wargocki et al., 2000). On the other side, increased ventilation rates, which can have some positive effects on indoor air quality (Seppänen et al., 1999, Seppänen et al., 2006), result in higher energy consumption. Ventilation heat losses/gains, due to differences in internal and external air temperature, directly affect both heating and cooling energy consumption and indirectly electricity consumption by requiring bigger fans to carry additional amount of air. All of this leads to larger HVAC system. On the other hand, the indoor air quality should also not be compromised by decreasing ventilation rates to conserve energy. In most countries nowadays, the minimum ventilation rate is specified by national regulations in order to provide a satisfactory level of indoor air quality. European Standard EN 13779:2007 (CEN, 2007a) classifies indoor air quality into four categories which are determined by the expected percentage of dissatisfied occupants (percentages presented in the European Standard EN 15251:2007 (CEN, 2007b):

- Category I: High indoor air quality (15% of dissatisfied occupants),
- Category II: Medium indoor air quality (20% of dissatisfied occupants),
- Category III: Moderate indoor air quality (30% of dissatisfied occupants), and
- Category IV: Low indoor air quality (>30% of dissatisfied occupants).

Typical range and default value of ventilation rate per person for each of these categories are presented in Table 2.6.

Table 2.6. Outdoor air ventilation rates [l/s per person]

Category	Typical range	Default value
I	> 15	20
II	10 – 15	12.5
III	6 – 10	8
IV	< 6	5

As mentioned earlier, the USA"s ASHRAE Standard 62.1-2007 requires minimum 8.5 l/s per person of fresh air to be secured in breathing zone for office spaces. According to the UK Building Regulations Approved Document F (ODPM, 2006a) a minimum of 10 l/s per person is needed to satisfy fresh air requirements in office buildings. This value falls between moderate and medium air quality according to the European Standard classification and it was used in this research to calculate the amount of outdoor air which has to be delivered to each zone in building models. Figure 2.11 shows the ventilation rate weekday daily pattern. It can be seen that the outdoor air is delivered to the building only during occupied hours (weekdays between 7am and 7pm).

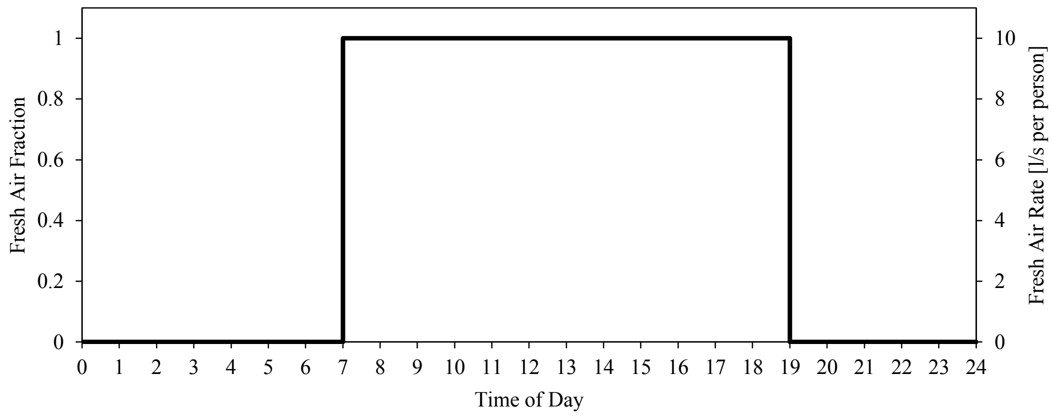


Figure 2.11. Ventilation rate weekday daily pattern

2.4.4. Internal heat gains

The main sources of internal heat gains are occupants, office electrical equipment and artificial lighting.

2.4.4.1. Internal heat gains from occupants

The density of occupation may vary considerably depending on the type of activity conducted in the particular office space. On the one side there are very high density spaces with the maximum density of 7 m^2 of floor area per person such as offices in an accounting section, while on the other hand the density in private offices may be as little as 19 m^2 per person or even lower for executives (ASHRAE, 2007).

The British Council for Offices (BCO) published the Occupier Density Study Summary Report (BCO, 2009a) which was based on detail occupancy survey in 249 UK commercial properties accommodating 173,000 workplaces in over 2,000,000 m^2 of Net Internal Area (NIA). NIA is defined as the usable space within a building measured to the internal finish of structural, external or party walls, but excluding toilets, lift and plant rooms, stairs and lift-wells, common entrance halls, lobbies and corridors, internal structural walls and car-parking areas (RICS, 2007). By claiming (with 99% confidence) that the given sample is a fair representation of the UK‟s commercial office stock, the BCO calculated the mean occupation density of 11.8 m^2 of NIA per workspace. The 77% of the sampled properties had an occupant density of 8-13 m^2 per workspace. Office buildings with the higher density, 5 to 7 m^2 per workplace, accounted for 5% while the 18% of sampled properties were in the 14 to 38 m^2 per workplace range. Overall, 25% of properties were occupied more densely than 9.2 m^2 per workplace, while 25% of properties were occupied less densely than 12.6 m^2 per workplace. Similar results were obtained when the sample was broken down into working floors. From 677 floors, 71% had occupant density of 7-12 m^2 per workplace, additional 6% were in the 5-7 m^2 per workplace range and 23% of the sampled floors were in the 14 to 20 m^2 per workplace range.

The occupant density varies depending on the space arrangement. For the open plan offices, the occupant density was set to a higher density of 9 m^2 per person. Cellular offices are usually shared by two or three people or sometimes are designed for single occupancy, which lowers occupant density to 14 m^2 per person. Common areas were designed to allow maximum occupant density of 9 m^2 per person, although when coupled with occupancy fraction, the actual density in common areas was lowered to one quarter during occupied period. Figure 2.12 shows the occupant density weekday daily pattern adopted here. From the same figure, it can be seen that for the considered office hours, between 7am and 7pm, the occupant density has a stepped increase and decrease for the first and last couple of hours respectively. A two-hour lunch period, between noon and 2pm, was used in which occupant density was decreased by one quarter. During the weekends and holidays, the occupancy was set to zero.

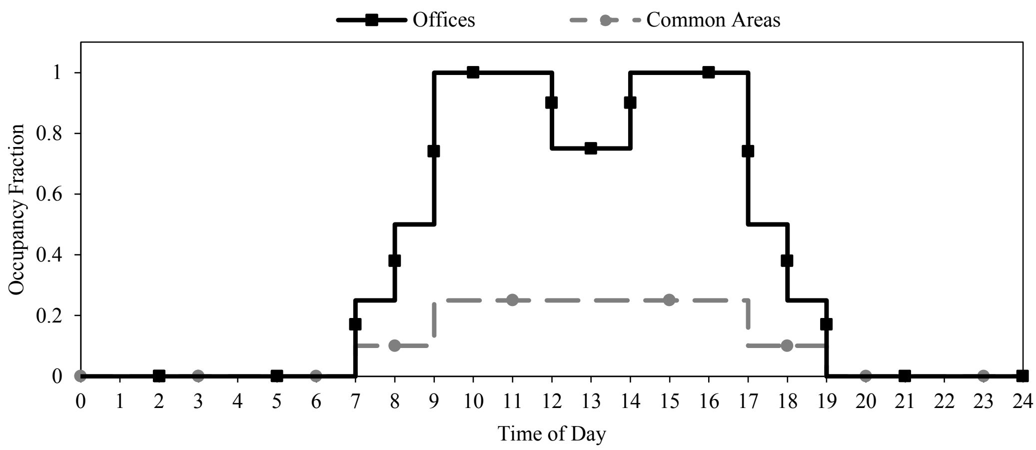


Figure 2.12. Occupant density weekday daily pattern (offices/common areas)

Occupant density is used to estimate a number of people in order to calculate heat gains from occupants. Each human, due to metabolic activities of the body, generates heat which is transferred to the environment. Amount of heat generated by a resting adult person and transferred through one square meter of skin is around 58 Watts and is called 1 met. The average male adult has a skin surface area of 1.8 m^2 which results in around 100 W dissipative heat from the resting adult person. Metabolic heat rate varies a lot depending on the activity level, person and surrounding conditions. ASHRAE Fundamentals (ASHRAE, 2009) gives the typical metabolic heat generation for various activities. Total heat generation is presented in Watts per person and adjusted to include both males and females. According to them, person who is doing very light work, seated, generates 115 W of total heat, of which 70 W is sensible heat. For the moderately active office work, the total heat production is 130 W (75 W sensible heat). Walking, or standing activity, which can be assigned to common areas, produces 145 W per person (75 W sensible heat). Similar value for the office sedentary activity can be found in the European Standard EN 13779:2007 (CEN, 2007a) which recommends heat production of 1.2 met (125 W) per person to be used.

In accordance with the ASHRAE suggestions and the European Standard EN 13779:2007 recommendations, the occupant total heat output was set to 125 W for "typical" office activity, while this value was increased to 145 W per person for the common areas. Figure 2.13 presents the metabolic heat rate together with the occupant density for the different area types which are used in building models.

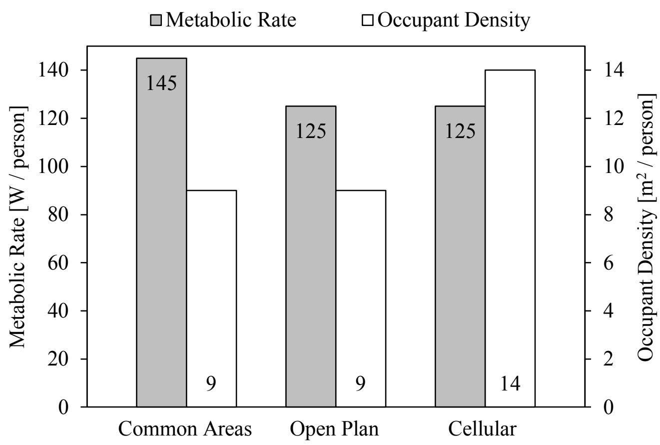


Figure 2.13. Occupant density and metabolic rate (offices/common areas)

2.4.4.2. Internal heat gains from office electrical equipment

Office electrical equipment, such as computers, printers, copiers, etc., can generate significant heat gains, which sometimes might be a dominant reason for implementing air-conditioning even in mild climates. However, the overestimation of office electrical equipment gains can result in increased capital and running costs of air-conditioning plant. Therefore, it is very important to carefully select values of equipment heat gains.

Office electrical equipment heat gains vary widely. Dunn and Knight (2005) calculated the internal heat gains of 30 air conditioned office buildings in the UK, based on surveys undertaken between April 2000 and October 2002. The results showed that the average equipment heat gains were 17.5 W/m^2 and ranged from 5.7 to 34 W/m^2. Once normalised for occupancy levels, the equipment heat gains had an average of 158 W per person and ranged between 124 and 229 W per person. They also reported strong correlation between equipment heat gains and occupant density.

In the more recent survey, undertaken by the Building Services Research and Information Association (BSRIA) and the British Council for Offices (BCO), sixteen buildings were studied between June and September 2008 (BCO, 2009b). From sixteen monitored buildings, three were excluded from analysis, which left thirteen office areas with total floor area of 24400 m^2. The reported average equipment heat gains of 13.9 W/m^2 and 140 W per person were slightly lower when compared to values reported in 2005 by Dunn and Knight. The reason for this might be the continuous increase in appliance efficiency. The highest value of heat gains was 36.2 W/m^2 while 85% of building had less than 25 W/m^2, and 70% had less than 15 W/m^2.

The office electrical equipment heat gains used in this research was calculated by multiplying the occupant density, which was set to 14 m^2 per person for cellular offices and 9 m^2 per person for open plan spaces, and previously mentioned average equipment heat gains of 140 W per person. This gives the equipment heat gains of 10 W/m^2 for cellular office areas and 15 W/m^2 for open plan areas. The latter value is equal to a typical office equipment load density in an open plan office stated in the Energy Consumption Guide 19 (ECG019, 2003). Figure 2.14 shows weekday daily pattern of office equipment usage and it can be seen that maximum equipment heat gains are generated during occupied hours. During unoccupied hours, it was assumed that 5% of equipment remains on, as it is often case in an office environment to have units which are in standby mode during night or required to be always powered. The graph at the right in Figure 2.14 presents the equipment heat gains of each particular space arrangement normalised per square meter. It can be seen that 2 W/m^2 were set for common areas which represents heat gains from equipment typical for those spaces such as reception desk equipment, vending machines, hand dryers in toilets, etc.

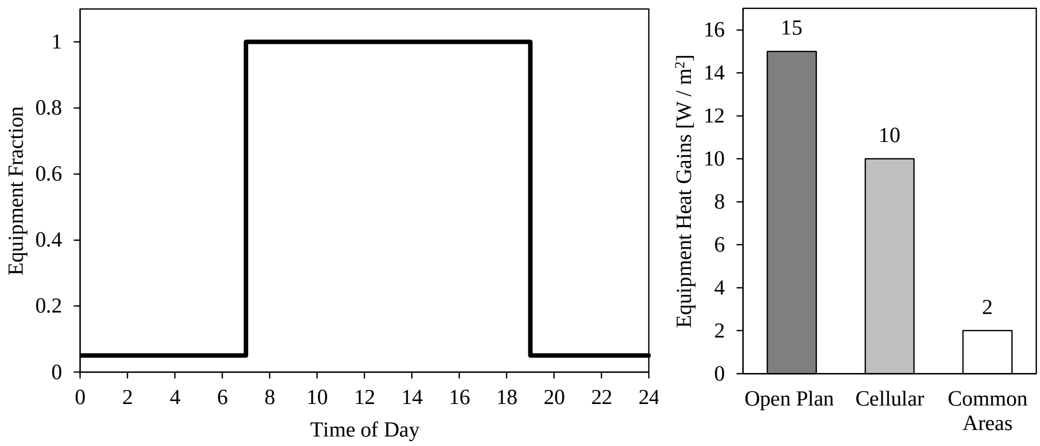


Figure 2.14. Office electrical equipment – weekday daily pattern and heat gains

2.4.4.3. Internal heat gains from artificial lighting

The artificial lighting is one of major electricity consumers in buildings. Installing the efficient lighting system is one of the most favoured strategies in reducing building electricity consumption. Reduced heat gains result in additional savings provided by lower cooling needs, although increased heating demand can lessen overall benefits. Zmeureanu and Peragine (1999) calculated that the net energy savings are about 70% of the gross lighting energy savings for a building located in a moderate/cold climate, while in a cooling dominant climate the net energy savings can be up to 20% higher (Sezgen and Koomey, 2000).

The main task of artificial lighting in office buildings is to provide sufficient quantity and quality of illumination so workers can perform the visual task efficiently and accurately without experiencing discomfort. The EU Standard EN 12464-1:2002

(CEN, 2002) gives recommendations in terms of maintained illuminance which is defined as *"the value below which the average illuminance on the specified area should not fall"*. The recommended values of maintained illuminance over the task area in any room where office work is carried out is generally in the range 300 to 500 lux, except for technical drawing where this should be 750 lux. Illuminances at the lower end of this range should be used for tasks such as filing, copying, telephone sales, etc. Where the tasks are mainly document based, such as writing, typing, reading and data processing, then 500 lux is required.

In addition to office areas, commercial buildings also accommodate so-called secondary office spaces which are the areas used to back-up the normal office work of the company. Lighting levels in these areas do not need to be high as in offices. According to the CIBSE Lighting Guide 7 (CIBSE, 2005b), illuminance of 200 lux is required for entrance halls, reception areas, tea points and rest rooms. Stairs and escalators should have lighting level of at least 150 lux, while recommended maintained illuminance of 100 lux at floor level is sufficient for corridors.

The installed lighting power density (W/m^2) depends on lamps efficiency as well as on control gear. Most areas can be lit by using no more than 2.5 W/m^2 of installed lighting power per 100 lux of maintained illuminance (BRE, 2004), which gives 12.5 W/m^2 for a 500 lux installation. Energy Consumption Guide 19 (ECG019, 2003) gave benchmark values for lighting power densities where the maximum value for offices is 12 W/m^2. This value is compatible with a maximum lighting power density allowed by ASHRAE Standard 90.1-2007 (ANSI/ASHRAE, 2007b).

The lighting power density level of 12 W/m^2 in office areas, which is set to comply with a benchmark value, is used in this research, as it can be seen in Figure 2.15. As common areas are composed from various spaces, for which illuminance levels very between 100 and 200 lux, the value of 3.4 W/m^2 was found sufficient to represent average lighting power in these areas. From the same figure can be seen that artificial lights were set to be operated only during the occupied hours.

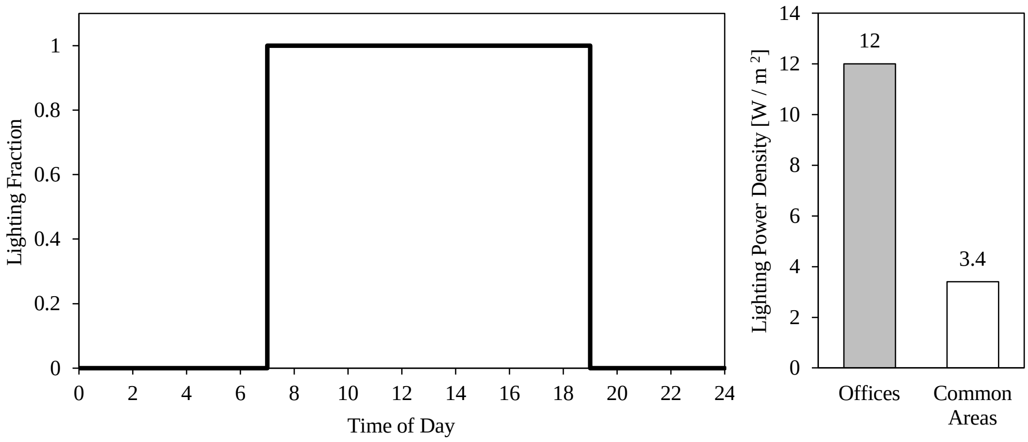


Figure 2.15. Artificial lighting – weekday daily pattern and power density

All the electrical energy used by artificial lighting system is ultimately released as heat which contributes to zone loads. The energy is emitted primarily by convection and radiation where the radiant part includes both long-wave (thermal) radiation and short-wave (visible) radiation. Fractions of convective and radiant heat in emitted energy depend on type of lamp and luminaire and the way luminaire is mounted. In the case luminaire is vented by connecting it to a return air duct, the fraction of the heat from lights would be removed by the return air stream and would not contribute to zone loads. Approximate values of return air fraction, fraction radiant, fraction visible and the fraction of the heat from lights convected to the zone air for different luminaire configurations and types are given in the IES Lighting Handbook (IES, 1993). For the purpose of this research it has been assumed that luminaires are not vented which limits return air fraction value to zero. Approximately 37% of heat is released by thermal radiation, while 18% is emitted by visible radiation. Rest of energy, which is around 45%, is convected to the air. Abovementioned values are typical for the overhead recessed fluorescent luminaires which are quite often used in office spaces.

2.4.4.4. Daylight control

Although internal heat gains can significantly contribute towards cooling loads and energy use for lighting can present a significant portion of buildings total carbon footprint, often artificial lighting is used with no control. There is no doubt that the daylight control can significantly reduce energy consumption of both cooling and artificial lighting. On the other hand, heating consumption might increase due to lower internal heat gains but not enough to diminish positive effect of a daylight control.

In order to inspect the influence of daylight control on building thermal behaviour, and to compare energy needs of buildings with and without daylight control, a new subset of building models were developed with daylight control implemented in office zones. For that purpose, the DElight simulation engine, integrated into EnergyPlus, was used. This particular model has been selected because it is, according to Carroll and Hitchcock (2005), robust, usable, and capable of providing sufficiently accurate quantitative information about performance of daylighting and lighting control systems in buildings. Maamari et al. (2006) did a validation study which compared several daylight simulation tools and came to the conclusion that DElight results correlated very well with the measurements.

The principle of operation is as follows:

- The DElight model calculates the interior daylighting illuminance level at user specified reference points and then compares it with illuminance target value, which was set to 500 lux for the office type activity.
- Artificial lighting is reduced whenever it is possible to benefit from daylighting while still achieving the desired target.
- The lights dim continuously and linearly from maximum electric power (maximum light output) to minimum electric power (minimum light output) as the daylight illuminance increases. Both, the minimum electric input power and the minimum light output were set to 10% of the maximum values.
- Once the minimum point is reached the lights stay on despite further increase in the daylight illuminance. This strategy was chosen to avoid frequent switching on and off which may occur during unstable weather conditions due to fast moving clouds.

The way in which the office zones in each building type are divided into several daylighting zones controlled by corresponding reference point is presented in Figure 2.16. The open plan office in building type one and building type three has five reference points, four perimeters and one core, while the open plan zone in the building type four has three reference points. The ability to include up to one hundred reference points per zone in its interior illuminance and artificial lighting reduction calculations was one of the main reasons why the advantage was given to the DElight model instead of the EnergyPlus native “Daylighting:Controls” method which has the limitation of maximum two reference points per zone. Cellular offices in building types two and four have only two daylighting zones, one close to the glazing area and another deep in the space.

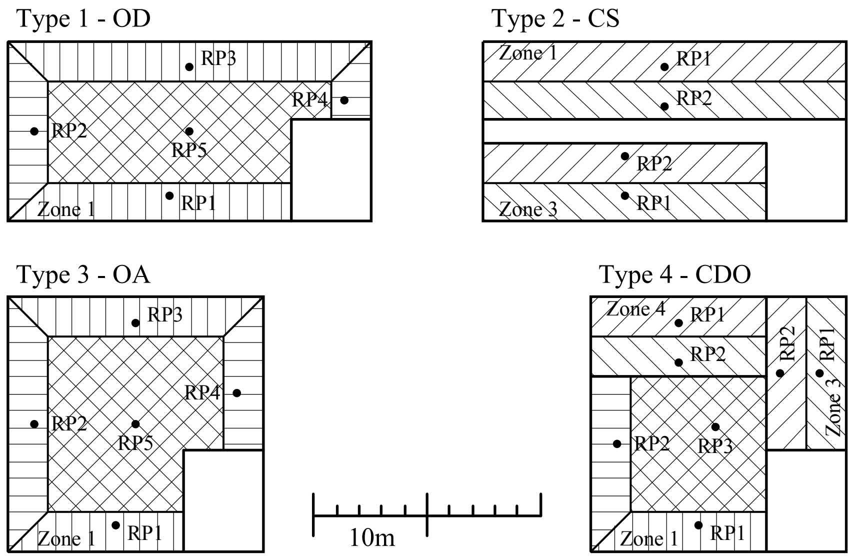


Figure 2.16. Daylight reference points

2.5. Office building model summary

This chapter has described various office building parameters used for the development of 3,840 office building models used in this research. Office building models have a basis in four the most typical office building build forms (Figure 2.8). These four build forms represent: open-plan sidelit buildings (OD); cellular sidelit buildings (CS); artificially lit open-plan buildings (OA); and composite sidelit cellular around artificially lit open-plan buildings (CDO).

In order to represent a range of existing buildings constructed during the past decades, as well as new buildings, each of these four building forms were coupled with five types of building fabrics. Building fabric type 1 (BF1) represents buildings with minimum or no insulation at all. Building fabric types 2, 3 and 4 (BF1, BF2 and BF4) represent buildings constructed according to Part L 1990, Part L 1995 and Part L 2002 Building Regulations respectively, while the building type 5 (BF5) represent buildings build according to the current UK Best Practice.

Besides the building form and fabrics, the size of the fenestration areas was also varied. Three levels of glazing ratio were included in the study: 25%, 50% and 75%. Furthermore, two measures of reducing solar heat gains were considered as design options: placing horizontal overhangs twenty centimetres above a window with a depth of 0.7 meters and replacing standard glazing with reflective glazing. Both standard and reflective glazing properties, such as solar heat gain coefficient and light transmittance factor, are presented in Table 2.4.

Daylight control was also implemented as a possible design option in order to reduce internal heat gains and artificial lighting electricity consumption. Finally, the orientation of buildings was also investigated by rotating the buildings at 45 degree intervals. Figure 2.17 presents all the possible scenarios which can be derived by combining these parameters.

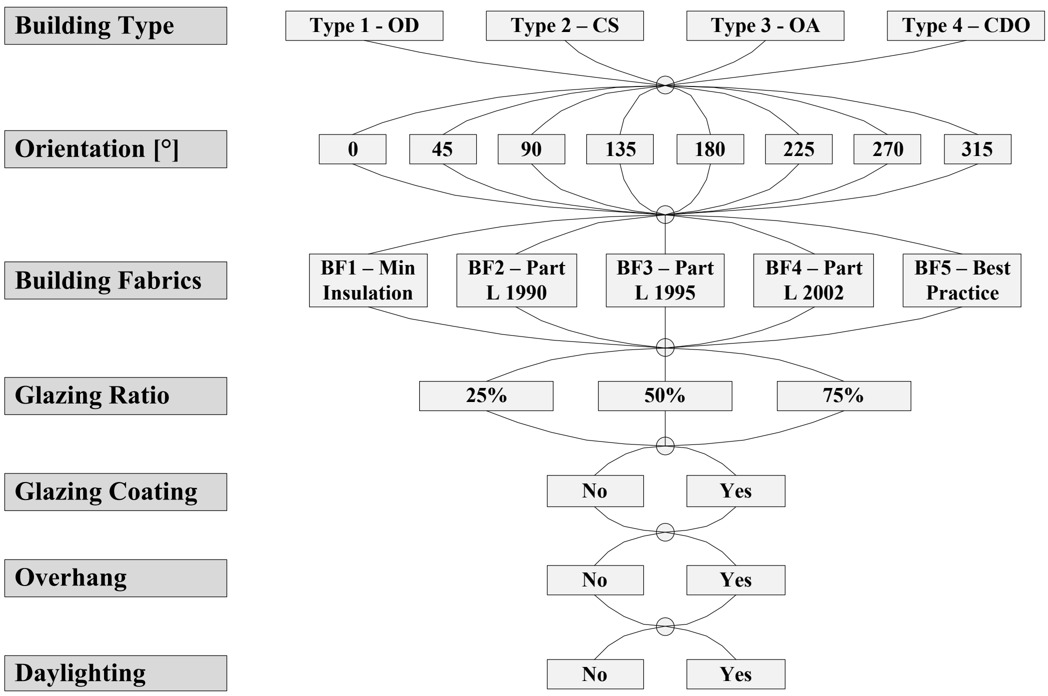


Figure 2.17. Office building models parameter tree

In addition to building parameters, the environment and activity related parameters differed according to building type. These parameters are summarised in Table 2.7 and include: zone temperature heating and cooling setpoints; fresh air ventilation rate; occupant density and metabolic rate; internal heat gains from office equipment; and internal heat gains from artificial lighting.

Table 2.7. Office building environment and activity related parameters

	Open-plan office space	**Cellular office space**	**Common areas**
Heating / cooling temperature setpoint [°C]	22 / 24	22 / 24	20 / 26
Fresh air ventilation rates [l/person]	10	10	10
Maximum occupant density [m^2/person] / metabolic rate [W/person]	9 / 125	14 / 125	2.25 / 145
Equipment heat gain [W/m^2]	15	10	2
Lighting power density [W/m^2]	12	12	3.4

Chapter 3

HVAC Systems and Simulation Models

Office building parameters, which are listed in the previous chapter, describe either the building envelope through parameters related to a building construction such as building shape, insulation level, glazing ratio, etc., or desired indoor conditions and building operation through parameters such as temperature setpoint, number of occupants, operating schedules, fresh air requirements, and other. These parameters as well as the type of heating, ventilating and air conditioning (HVAC) system that the building is coupled with affect building energy requirements, both thermal and electrical. The function of an HVAC system is to provide and maintain satisfactory artificial environment conditions, for the comfort and welfare of the occupants. Full year-round HVAC system operation provides simultaneous control of room dry-bulb temperature, relative humidity, fresh air requirements, air purity and motion, and room sound level. This is achieved by simultaneous ventilation and either heating or cooling (or both if there is inequality in requirements). Humidification or dehumidification processes can be applied in cases when humidity has to be controlled in narrow range.

This chapter firstly presents the classification of HVAC systems and briefly describes basic characteristics of various HVAC system types. The systems description is followed by a comprehensive literature review on the energy efficiency of different HVAC systems and the systems distribution across an office building stock. Last part of the chapter is focused on the detailed description of HVAC system simulation models developed for the purpose of this research.

3.1. Classification of HVAC systems

Due to the large number of variations in HVAC equipment and the ways they can be used to control an indoor environment on one side, and on the other side different energy requirements in buildings, HVAC systems are designed and operated in different ways and assembled in various configurations. The main aims of classification of HVAC systems are to make a distinction between the variety of types and to provide a background for selecting the most suitable system based on building requirements (Marjanovic-Halburd et al., 2008). HVAC systems are usually firstly divided into two categories: individual and central systems. Individual systems use self-contained, factory-assembled units with a main purpose to serve one room only, although sometimes two rooms can be conditioned, and typical examples are widow air conditioners and DX split-type systems. Central systems are composed of two main parts: a primary system and a secondary system. The task of the primary system is to generate energy which is distributed via a secondary system to individual zones in order to maintain desired indoor conditions. The usual approach in building services textbooks and design manuals is to classify central systems based on their working fluids as all-water, all-air, air-water and systems which use refrigerants. The basic HVAC system classification is presented in Figure 3.1. Principles of operation and fundamental characteristics of various air conditioning systems are the main topics in building services and HVAC handbooks (CIBSE, 2005a, ASHRAE, 2008, Kreider, 2001, Pita, 2002, Eastop and Watson, 1992). Following sections of this chapter briefly summarise the basic of functioning of the most typical HVAC systems.

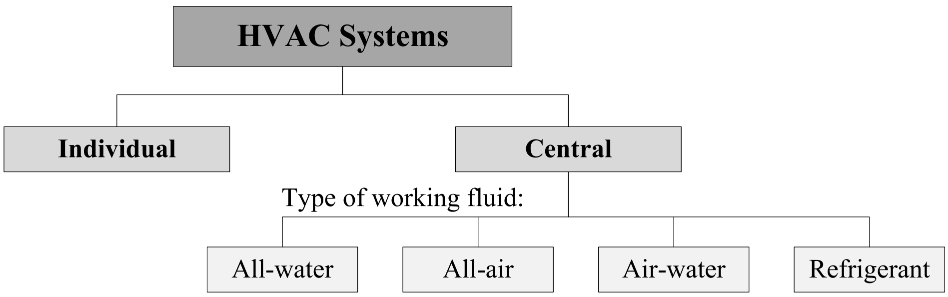


Figure 3.1. HVAC systems classification

3.1.1. All-water systems

The basic concept of all-water systems, which are frequently called hydronic systems, is based on continuous distribution of hot or chilled water from a central plant to hydronic terminal units located in individual zones, as presented in Figure 3.2. Hydronic terminal units are heat exchangers which transfer the energy between the air in the space and the circulating water inside the unit. They can be divided into two categories: units that are suitable for heating only and units which can be used for both heating and cooling. Typical heating only units are radiators, convectors, baseboard units, finned-tube units and unit heaters, while fan-coils and radiant panels are typical representatives of both heating and cooling equipment.

The main advantage of all-water systems is that they require less building space than all-air systems as there is no need for air ducts and central air handling equipment. This is particularly important when space is extremely limited which is often a case in buildings originally designed not to be air-conditioned. This makes all-water systems a suitable choice for refurbishment projects.

On the other hand, the main disadvantage of this system type is very limited ventilation as well as humidity control. These systems have no central air distribution which makes them difficult or even impossible to fulfil ventilation requirements. In most cases, they are coupled with natural ventilation strategies, although there are some types of fan-coil units which allow outdoor air to be brought directly into a zone via opening in the rear. In this case, the unit has to be placed along an exposed wall that has an opening for a fresh air connection.

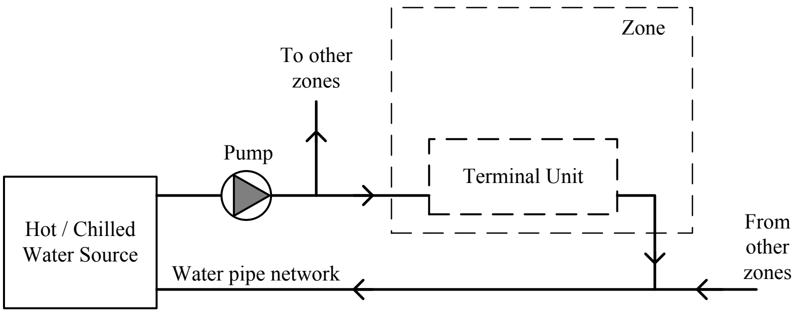


Figure 3.2. All-water system

3.1.2. All-air systems

An all-air system transfers the air via duct network from a central plant to the zone at such a condition that, when mixed with the air in the room, brings the room condition to the desired level. Complete air preconditioning, which include preheating, reheating, cooling, dehumidification/humidification, happens before the air is delivered to the room, leaving the room air without any further need for a treatment.

All-air systems have many advantages. They offer a great flexibility in temperature and humidity control. In addition, energy saving measures may be easily incorporated in these systems, such as the usage of an air-side economizer to increase free cooling capabilities or the installation of air-to-air or other types of heat recovery units. Potential harm for occupants and furnishing is reduced as potentially dangerous electrical equipment, wiring and piping are kept away from occupied zones. In addition, the major air conditioning equipment, such as pumps, fans and air handling units, are located in the plant room, which allows operation and maintenance to be performed in unoccupied area. Beside many positive features of all-air systems, there are some disadvantages too. Usable floor area is reduced by allocating additional space for vertical shafts required for air distribution. Beside the larger floor area requirements, the building height has also to be increased as air ducts are mostly placed in suspended ceiling. In addition, the balancing of all-air system may be more difficult, particularly on large systems.

All-air systems are usually classified into three categories:

- Single duct systems,
- Double duct systems, and
- Packaged systems.

Single duct systems may be further divided into two categories: constant air volume (CAV) systems and variable air volume (VAV) systems, while double duct systems are subdivided into dual ducts and multizone systems. Dual ducts systems are additionally subcategorised into constant air volume systems and variable air volume systems. The all-air systems classification is presented in Figure 3.3.

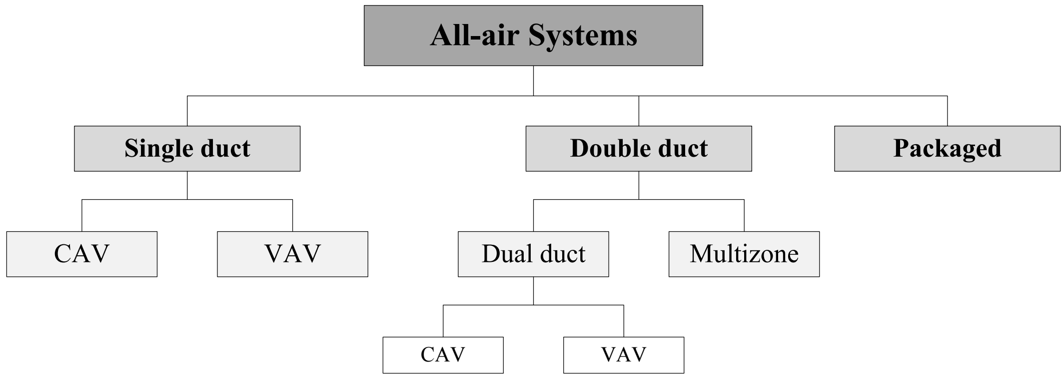


Figure 3.3. All-air systems classification

3.1.2.1. Single duct systems

In a single duct system, conditioned air is delivered to the zone by a single duct. The main heating and cooling equipment (hot/chilled water coils, gas furnace, etc.) are arranged in a series flow air path. Since the heating or cooling demand varies with time due to changes in outside temperature, solar radiation, occupancy, and other causes, the mechanism of adjusting the amount of energy delivered to the zone is necessary. Two basic types of such mechanism are the most common in single duct all-air system applications. First is to vary the supply air temperature while keeping the amount of supply air constant (constant air volume systems) and second is to modulate supply air volume while maintaining constant supply air temperature (variable air volume systems).

A single duct constant volume system varies the supply air temperature in response to the zone heating/cooling requirements while maintaining a constant air volume. The simplest form of this system is a single-zone system which distributes air to a single-temperature control zone. Typical applications include spaces with uniform loads, such as large lecture theatres or halls, and small spaces which require precise control, for example an operating block in hospital. This system is very inefficient in operation and provides very poor conditions if it is used for multiple zones with unequal loads. A modification of a single-zone system, which is much more suitable for applications with unequal loading, is a multiple-zone reheat system. Unequal loading often happens when system serves several perimeter areas with different exposures. The ultimate case is when system has to provide cooling in one part of a building while at the same time some zones require heating. Multiple-zones reheat systems deliver preconditioned air at a temperature sufficiently low to treat a zone with the highest cooling demands (or the smallest heat losses). For all other zones, which would be over-cooled, reheaters located in the supply ductwork to the each zone reheat the supply air. This result in a very high energy use as complete air volume has to be always completely cooled to certain temperature and then often reheated. Figure 3.4 schematically presents a constant air volume multiple-zone reheat system. A single-zone system would be represented if reheaters were excluded from the graph.

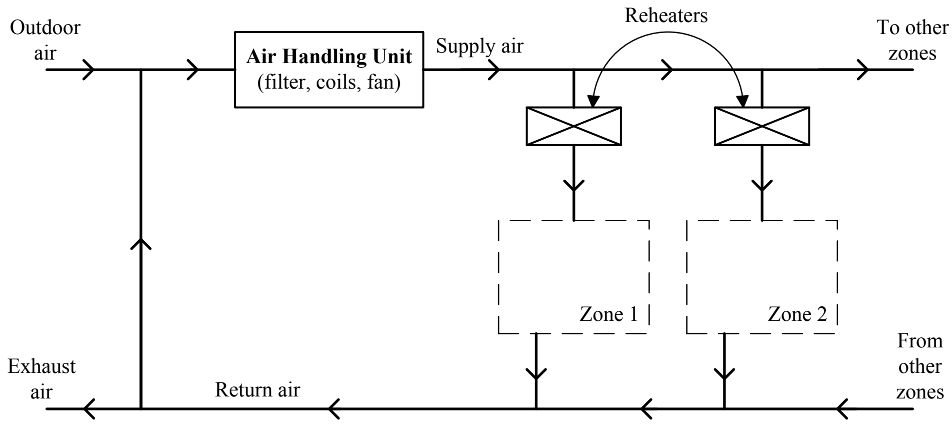


Figure 3.4. Single duct constant air volume multiple-zone reheat system

A single duct variable air volume system (Figure 3.5) controls the zone temperature by varying the quantity of supply air rather than the air temperature. The supply air temperature is kept constant, although can be adjusted seasonally. Preconditioned supply air is delivered to individual zones through air terminal units. Air terminal units control the air flow rate in response to the zone demand. These units are available in several configurations. The simplest one is throttling box, which is essentially an air valve or damper, which reduces the supply air flow in response to decreasing zone temperature. This unit is suitable for cooling only applications. To meet any heating requirements, air terminal units have to be equipped with reheaters, which are either hot water or electric. Induction or fan-assisted air terminal units are used when there is potential issue with air stagnation or low air movement. Variable air volume systems, due to reduce air volume flow rate during part load operation, can have potential problems with control of air humidity, air mixing within a zone, and securing sufficient amount of outdoor air. On the other hand, these systems have some advantages too. They conserve energy by running fans at a reduced volume whenever possible.

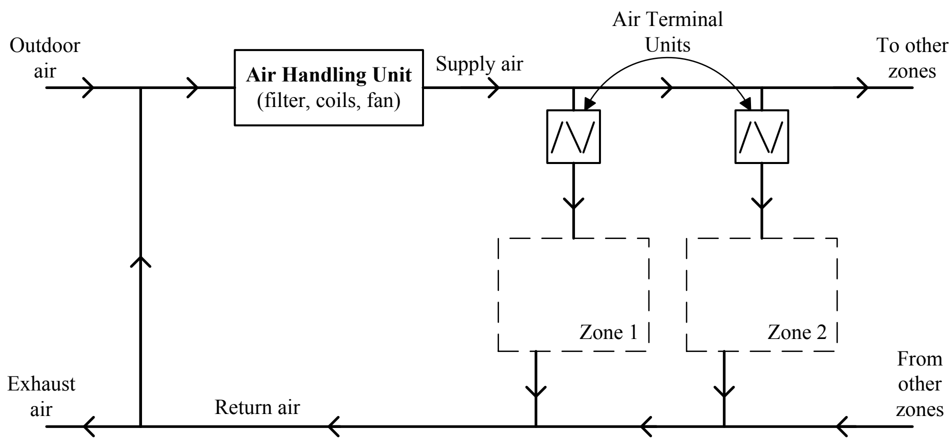


Figure 3.5. Single duct variable air volume system

3.1.2.2. Double duct systems

In contrast to a single duct air system, a double duct system contains the main heating and cooling equipment in parallel flow air paths. Air is delivered from an air handling unit to air-conditioned zones either by two separate cold and warm air streams which are blended in zone mixing terminal units (dual duct systems) or by a separate single duct to each zone where the supply air is mixed to the required condition at the main air handling unit dampers (multizone systems). The amount of energy delivered to a zone is usually controlled by varying supply air temperature; although in some applications the air volume flow rate is also modulated.

Dual duct systems (Figure 3.6) employ two ducts which separately supply cold and warm air to a zone mixing terminal unit where two air streams are mixed in appropriate proportions to respond to the zone demand. These systems can be operated as constant air volume and variable air volume. Constant air volume dual duct systems are similar in operation to single duct multiple-zone reheat systems with two differences. Firstly, air is reheated at a central plant rather than at individual zones. Secondly and more importantly, as it preserves energy, is that only part of air stream is cooled instead of cooling the entire volume of air and then reheat it to match the zone demands. The supply air temperature of each air stream leaving the air handling unit is controlled to provide minimum heating and cooling to satisfy a zone with the highest heat losses/gains. Variable air volume dual duct systems have several variations in design and operation. It can operate either by varying flow rates of both cold and warm air streams or by maintaining one air stream at constant flow rate. Blended air is delivered to zones through variable air volume mixing terminal units.

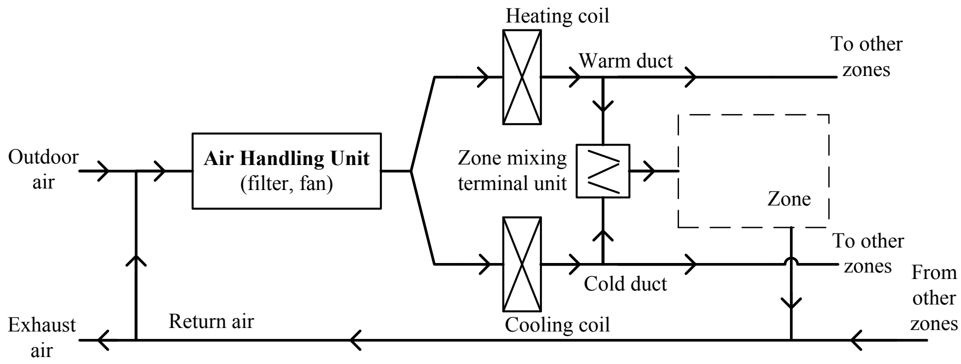


Figure 3.6. Dual duct air system

Multizone systems, which are often called hot deck/cold deck systems, are in principle very similar to dual duct systems with a major difference that air mixing occurs at the leaving of the central air handling unit instead at zone mixing terminal units. The mixed, conditioned air is delivered to individual zones by set of single ducts. This arrangement limits the number of zones which can be served from a single, centrally located air handling unit to up to twelve. Multizone systems can provide smaller buildings with advantages of dual duct systems while using simplified packaged equipment.

3.1.2.3. Packaged systems

Packaged systems are often classified as a subgroup of central all-air systems due to their similar operation. The major difference between packaged and central systems is that the former provide heating/cooling from their own equipment located in package unit. A packaged unit is always equipped with direct expansion coils for cooling which is the primary difference between a packaged unit and an air handling unit used in central air-conditioning systems, which is usually equipped with water coils. Heating in packaged systems can be provided in several ways: by gas-fired furnace, electric heaters, or air-source heat pump. The refrigerant plant is part of packaged unit, although condensing unit can be sometime placed externally.

Packaged systems provide minimum ventilation rates and can control humidity precisely, which is the same as in central HVAC systems. On the other hand, packaged systems cannot be coupled with complex duct network due to limited fan power. Heating/cooling energy is delivered to zones by a single duct and is controlled either by varying the supply air temperature while keeping the amount of supply air constant (constant air volume packaged systems) or by modulating supply air volume while maintaining constant supply air temperature (variable air volume packaged systems).

3.1.3. Air-water systems

Air-water systems exploit the best features of all-air systems and all-water systems. Preconditioned supply air, hot water and chilled water are distributed from a central plant to individual zones in order to perform heating and cooling function. Supply air usually operates with a constant air flow rate sufficient only to satisfy fresh air requirements, which means that in most cases 100% fresh air is delivered to conditioned zones. This is the reason why the air side of the air-water systems is also called a primary air or dedicated air. Applications with a supply of a fresh air only offer great opportunity for energy saving by implementing a heat recovery unit.

Although a part of energy requirements can be offset by the air side system, the majority of cooling and heating needs are covered by the water system via hydronic terminal units installed in each zone. Most popular hydronic terminal units which can provide simultaneous heating and cooling are induction units and fan-coils. Sometimes the energy is delivered to a zone by separated units, for example chilled ceiling for cooling and radiators for heating.

Air-water systems have several advantages in comparison to all-air systems. They require less building space as the major part of energy is delivered by water. Supplying the fresh air only also decreases the space for duct network. Another advantage is that individual zone temperature control allows the adjustment of each thermostat for a different temperature, which gives users more control over indoor conditions. On the other hand, air-water systems require more maintenance which, for a water side, happens in occupied areas. In addition, hydronic terminal units which operate in cooling mode at low dew point temperature need to be equipped with either condensing water network or condensate panes which have to be cleaned regularly. Another disadvantage is that due to operation with minimum air volume, the humidification and dehumidification capacity of air-water systems is limited.

3.1.4. Refrigerant systems

Refrigerant central systems are relatively new when compared to conventional HVAC systems. They provide cooling and heating by using refrigerant as a working fluid. An example of these systems is variable refrigerant flow systems (VRF), also called variable refrigerant volume systems (VRV), which is composed of two major parts: outdoor units and indoor units (Figure 3.7). The term Variable Refrigerant Flow refers to the ability of the system to control the amount of refrigerant flowing from outdoor unit to each of indoor units, enabling in that way the usage of various types of indoor units with different capacities. There are two basic types of VRF systems: cooling/heating only and heat recovery systems. Cooling/heating only systems provide just cooling, or cooing or heating if the heat pump mode is incorporated, however not both at the same time. Heat recovery systems provide simultaneous heating and cooling by diverting the heat extracted from zones which required cooling to areas which have heating demand.

The outdoor units used in VRF systems are air-cooled condensers. They are available with different cooling and heating capacities, and can be packed together to satisfy various building requirements. Each zone energy requirements can be met by the wide range of indoor units, which can be wall, floor or ceiling-mounted, or placed in ducts.

The flexibility of coupling a range of indoor units with properly sized outdoor units makes the VRF system an appropriate solution for a range of different building types. Although the fresh air supply is limited, requirements for most applications can be satisfied. The main disadvantage of VRF systems is decreasing the coefficient of efficiency when operations furthering away from standard rated conditions. For example, when the outdoor air temperature is low, which means that heating demand is high, the VRF system operates with the lowest efficiency. Another disadvantage is the quality of filters used in the indoor units that is lower than in conventional air handling units, which may affect the zone air purity.

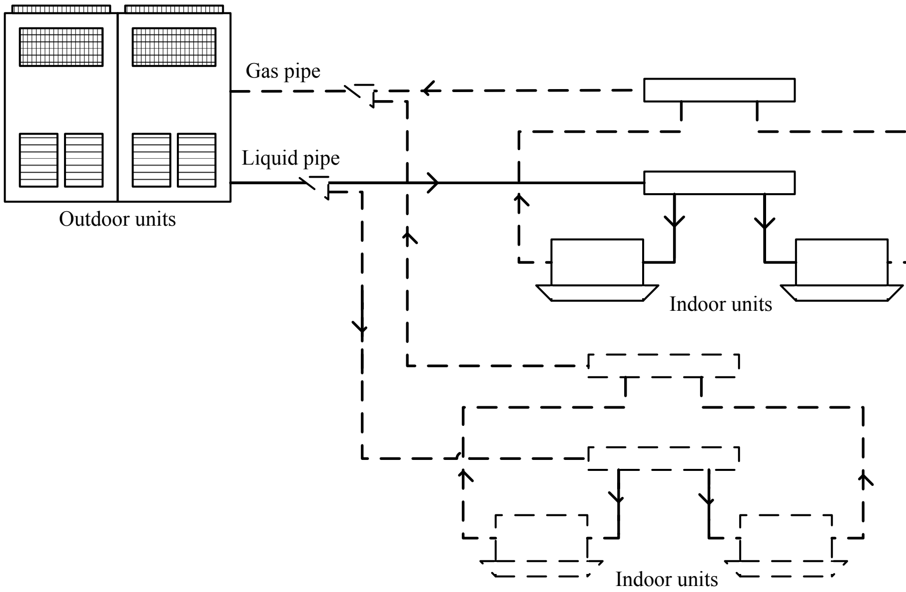


Figure 3.7. Variable refrigerant flow system

3.2. Energy performance of different HVAC systems

This HVAC systems classification shows that there is a wide range of air conditioning systems available. The energy performance of a HVAC system is determined by its design and suitability to meet the heating and cooling demand of a building. HVAC systems have been studied thoroughly during last several decades and there are many publications available which describe their performance and compare the efficiency of different systems.

Aktacir et al. (2006) compared a variable air volume system (VAV) and a constant air volume system (CAV) based on the life-cycle cost analysis by using building energy requirements as well as initial and operating costs. They made a building model which was coupled with two activities, school and office, and two operating hours, between 8am and 5pm and between 8am and 12pm. The initial costs of HVAC equipment in the case of the school building was about 7% higher than that of the office building, mainly due to requirements for a larger chiller. When compared the initial costs of VAV and CAV systems for the same building, they found that the initial cost of the VAV system was approximately 22% higher due to more complex control units and additional VAV terminal boxes. Another conclusion was that the operating cost of the VAV system is significantly lower when compared to the operating cost of the CAV system. The annual saving was affected by the operating time in the way „more system is operating, the savings are higher". For the school building, the total operating saving of the VAV system was 21% and 25%, depending on the operating hours. The savings for the office building are even higher, 24% and 30%. The highest impact on such a significant savings had the fan electricity consumption, which was in the case of the CAV system up to 60% higher. Authors also calculated the payback period and found that, for the long operating hours, the VAV system is an excellent alternative to the CAV system as the payback period is less than four years. However, for the short operating hours, where the payback period is more than ten years, the VAV system is not a very attractive choice.

Kalaiselvam et al. (2006) did a comparative study on energy performances of the VAV system and the CAV system when serving a software laboratory. High internal heat gains from occupants (165 people) and computers (133 W/PC station) mainly drove cooling requirements of the software laboratory. They observed the occupancy and found that there were two peaks, one morning and one afternoon, when the laboratory was maximally occupied. Rest of the time, the occupancy was between 30% and 70%. This was important because the amount of outdoor air supply to the space in order to fulfil fresh air requirements was controlled by CO_2 concentration. By comparing the performance parameters of two studied systems, which included chiller energy consumptions and fan power consumptions, they reported that the total energy saving of the VAV system were between 15% and 22%, depending on the cooling load. In addition, the chiller required between 9% and 15% less energy, while the fan power consumption was lower between 10% and 45%.

Potential energy savings of the VAV system when compared to the CAV system in hot and humid climates was investigated by Sekhar (1997). He prepared five building models, three with a square floor plan and the rest with the rectangular shape, with a different orientation and a percentage of glazing. Simulation results showed that the space cooling energy savings ranged between 11% and 19% in favour of the VAV system, while savings in the fan energy amounted between 50% and 70%. Depending on the building characteristics, total energy savings of the VAV system could be from 11.5% up to almost 26% in buildings with the large amount of glazing. Yang and Ting (1999) set up a full-scale experiment in which measured energy consumption of two test cells, one equipped with the CAV system and another with the VAV system. According to them, energy savings of the VAV system in hot and humid climate can be up to 50%, while the minimum expected savings should be around 30%. They also reported that the VAV system had difficulties in achieving same space humidity as the CAV system due to inadequate dehumidification capacity at the part load operation.

Yao et al. (2007) extended the analysis by including a fan-coil system in a comparison. They simulated a small office building by using six weather files which represent different climate regions in China. Simulation results showed that the most energy efficient system was the VAV followed by the fan-coil system, while the least efficient was the CAV system, independent of weather data. The VAV system, as the best, was able to reduce total energy consumption by between 17% and 28% when compared to the CAV system, and between 4.6% and 10% in comparison with the fan-coil system. The savings were higher in warmer climate regions. Sekhar and Yat (1998) chose five different types of HVAC systems for a simulation study of a large twenty-storey office building with a 60 meters by 40 meters foot-print. Beside the VAV system, CAV system and two pipe fan-coil system, they also investigated the performance of packaged variable air volume system and two pipe induction unit system. From the simulation results, they found that the two pipe induction unit system has the lowest annual energy consumption; 5% less than the energy consumption of the VAV system, which was set to be the reference system. All other systems performed worse than the reference system. The CAV system consumed about 10% more energy, while the most inefficient was the packaged system which required slightly below 20% more energy than the VAV system. The fan-coil system performed similarly to the CAV system, mainly due to high energy consumption of unit fans, although authors claimed that this system could be more energy efficient than the VAV system if fan-coil units fan energy consumption is better managed.

Better performance of a fan-coil system, when compared with the VAV system, has been claimed in the simulation study done by Zhou et al. (2007). The estimated energy saving potential of the fan-coil system was around 10%. However, the main aim of abovementioned study was to compare the energy consumption of the variable refrigerant flow system (VRF) with two conventional HVAC systems, in particular the VAV system and the fan-coil system. An EnergyPlus model of a ten-storey multi-zone building was developed and simulation outputs for each of systems were compared. The overall conclusion was that the VRF system is an energy efficient alternative to conventional systems as it can achieve up to 23% and 12% total energy savings when compared to the VAV system and fan-coil system, respectively. A similar simulation comparison of VAV and VRF systems was done by Aynur et al. (2009). A VRF system, equipped with heat recovery ventilation system which provides fresh air, was compared to a VAV system. It was found that using the VRF system instead of the VAV system could reduce total energy consumption between 27% and 58%. The scale of energy savings depends on the type of VAV boxes used in the VAV system. In the case where VAV boxes with no reheat units were used, the potential saving was smaller. However, the VAV system with no reheat was not able to properly maintain the desired indoor temperature. By installing reheaters the temperature was maintained properly, although with significant energy penalties.

There is no doubt that traditional constant air volume systems (CAV) are highly inefficient. Unfortunately, this system type was very popular in 1950s and 1960s and many installed systems are still in use due to high replacement costs (Cho et al., 2009). This is the reason why several strategies of how to improve these systems were developed. Cho and Liu (2009b) presented a procedure for supply fan speed control by installing variable frequency drive (VFD) on existing CAV systems. They tested a procedure on the 12-storey office building built in late 1960s. The most important step in the procedure was to determine an optimal fan speed based on the highest zone load. After applying the optimal supply fan control method, they measured and analysed zones conditions and energy consumptions. Implementation of the VFD showed tremendous energy saving without compromising indoor thermal conditions. Fan electricity consumption reductions were between 60% and 75%, depending on the particular zone loads, while the reheat energy requirements were reduced by 43%.

Overall, the total electricity savings were around 23% and gas savings almost 19% over a six-month monitoring period. Another way of increasing energy efficiency of a CAV system in retrofit projects is to install VAV terminal boxes while keeping the same basic air-side equipment. Decreased airflow rate during part load operation reduces energy requirements for reheating. Ardehali and Smith (1996) reported the payback period of only 3-4 years for office buildings by implementing this strategy, mainly due to significant savings in electricity and gas consumption. The best results in reducing energy consumption of a CAV system by modification can be obtained by combining previous two strategies: to install VAV terminal boxes and modulate the airflow with variable frequency drive on air-handling units. In this way, the operation of a CAV system would be very similar to a VAV system.

A relatively new system, which has attracted increasing attention in recent years and has certain potential to be energy efficient alternative to traditional HVAC systems, is the chilled ceiling system. Chilled ceiling system with a fresh air ventilation may or may not reduce energy consumption which depend on a lot of parameters such as climate conditions, supply air temperatures, outdoor airflow rate and cooling loads (Novoselac and Srebric, 2002). Sodec (1999) compared the energy consumptions and energy costs of a VAV system and chilled ceiling system based on simulations. The chilled ceiling system had around 12% higher total energy consumption for the specific cooling load of 46 W/m^2. For the 56 W/m^2 of specific cooling load, the difference was only 6%, while the chilled ceiling system became slightly more efficient at the 75 W/m^2, when compared with the VAV system. Similar comparison based on simulation was performed by Niu et al. (1995). The results indicated that for office buildings in temperate climates a chilled ceiling system has a compatible energy performance to a VAV system, and up to 25% lower energy requirements in comparison with a CAV system.

Tian and Love (2005, 2009) analysed the performance of radiant cooling-VAV system combination in a constructed university building in Canada. They found that in a cold and dry climate, a conventional VAV system could have at least 30% lower annual energy consumption than installed radiant cooling system. This appeared mainly due to simultaneous heating and cooling which occurred in the existing system and a great potential for free cooling which can be effectively used in a VAV system. To investigate possible ways of improving the existing system, they created EnergyPlus models of the particular building and HVAC system, which were calibrated by measured heating and cooling energy use, as well as lighting and equipment electricity consumption. In addition, occupancy, lighting, appliances and HVAC system operation schedules were adjusted to represent real observed conditions. By simple control modifications, they succeed to match energy consumption of a VAV system. Modifications included heating setback temperature reduction, cooling setpoint increment by 1°C, and switching off chilled panels between October and April when the cooling demand can be met by outdoor air. Further measures, which included modification of existing system configuration and a better control of solar gains and heat losses through changes in building fabrics, resulted in significant savings. The radiant cooling system could achieve up to 80% savings in total annual energy consumption in comparison with the conventional VAV system.

On the other hand, Knight and Dunn (2005) claimed that chilled ceilings are the most efficient system for HVAC applications in UK office buildings. This statement was based on findings from a two-year field energy monitoring study of air conditioning systems in 32 UK office buildings. Beside five buildings, which had chilled ceilings, other monitored buildings were equipped with different HVAC systems including: all-air systems, either VAV or CAV, two-pipe or four-pipe fan-coil systems, VRF systems or direct expansion systems (single or multiple systems). Annual energy consumptions and carbon emissions were compared among different systems as well as with a UK good and typical practice benchmarks. Good and typical practice benchmarks are given in the "Energy Consumption Guide 19: Energy Use in Offices" (ECG019, 2003). All buildings with chilled ceilings were quite below good practice benchmark values. VRF systems also performed significantly better than good practice standards. The majority of traditional HVAC systems (all-air and fan-coil systems) met or exceeded current typical practice standards, while more than half of them exceeded current good practice standards.

Zhang et al. (2006) used a different approach in assessing an energy performance of different HVAC systems. They presented an HVAC system as a set of basic individual components connected by airflows which can be evaluated by using psychrometric and energy balance analysis. For the purpose of research, ten possible two-zone HVAC system configurations were developed, including single-duct, dual-duct and fan-coil based systems. The performance of each of these systems was evaluated by comparing their energy consumptions at different operating conditions. Test cases included a variety of outdoor and indoor conditions, minimum fresh air requirements, and zone sensible and latent loads. Main findings from the analysis were that the optimal system, which means the system with minimum energy consumption, need to meet the following criteria:

- to minimise ventilation losses, while maintaining satisfactory indoor air quality by fulfilling minimum fresh air requirements,
- to eliminate simultaneous heating and cooling, e.g. chilled air from the main AHU reheated in zone reheaters,
- to maximise the usage of a free cooling whenever possible, and
- to be able to exchange energy between different zones with cooling and heating demand either by implementing inter-zonal airflow or heat exchanger.

Such a system should be based on fan-coil units, equipped with a dedicated air system, and have inter-zonal airflow paths. However, the question remained, how difficult is it to install such a system in real buildings, if possible at all.

Wright and Zhang (2008) noticed that a current practice in evaluating HVAC systems is either to compare system performance with performances of similar systems or to use life-cost analysis. This approach does not answer the question how far particular system consumption is from the optimal system performance, so they introduced the term “system effectiveness”, which was defined as a ratio between the minimum possible system capacity and the actual system capacity. Minimum system capacity offset ventilation and zone thermal loads and maximise positive effects of inter-zonal energy transfer. The concept was proofed on the example composed of five thermal zones for which the minimum system capacity was calculated by using optimisation. A conventional HVAC system, which was coupled with a five-zone example and for which the effectiveness was calculated, was based on the VAV system.

The total energy consumption of such a system was more than 60% higher than the optimal system consumption, which resulted in effectiveness of only 0.61.

Besides the design characteristics, the control and operation of HVAC systems also have significant impact on their efficiency. Advanced control strategies, such as supervisory and optimal control, can improve the performance of HVAC systems. Lu et al. (2005a, 2005b, 2005c) found that the HVAC system can have energy consumption reduced by more than 10% if its components are operated at the optimum setpoints instead of traditional control strategies. Optimal setpoints include chilled water temperature setpoint, chilled water pump head, pressure drop in duct network, and the sequencing of chillers and pumps. Kusiak et al. (2010) demonstrated that the total energy consumption of the VAV system can be reduced by more than 7% if the supply air temperature and the static pressure in the air handling unit are set to an optimal value.

Wang and Ma (2008) presented a comprehensive review of supervisory and optimal controls of building HVAC systems. This review covered major findings from a large number of state of the art research and development studies, as well as applications of supervisory and optimal control. Their general conclusion was that the implementation of advanced control strategies in building HVAC systems has a high energy/cost saving potential. However, the majority of proposed supervisory control strategies were validated only by simulations or by pilot tests on small-scale HVAC systems, while the practical validation of large and complex HVAC systems is still missing. This led them to conclusion that more research and development is needed in order to make supervisory and optimal control functional in real applications. On the other hand, Chapter 41 of the ASHRAE HVAC Applications Handbook (ASHRAE, 2007) presented a number of near-optimal control strategies, which are simple and applicable in practice. Some of these are: strategies for air handling units, cooling thermal storage control, cooling tower fan control, chilled water reset with fixed and variable speed pumps, sequencing and loading multiple chillers plant, etc.

Another measure that might have an impact on HVAC system energy consumption is the usage of a night ventilation precooling. Braun and Zhong (2005) analysed the impact of night ventilation precooling on a range of buildings in the simulation study. They found that annual savings in compressor energy could be as high as 53%, depending on a location and building type. However, the total energy savings were much smaller, up to 8%, due to increased fans electricity consumption. Kolokotroni et al. (1998) assessed the suitability of night ventilation to cool UK office buildings and found that it is a valuable method to address the issue of summer overheating. Kolokotroni and Aronis (1999) found that application of night ventilation results in an energy saving of at least 5% for typical UK office buildings. Potential saving can be increased to about 15% for the case of a heavyweight building. Braun (2003) provided an overview of researches related to the impact of night ventilation on HVAC system energy consumption. The overall conclusion was that the night ventilation can provide significant savings, but it is highly sensitive to many factors such as type of equipment, building constructions, occupancy schedule, climate conditions, and control strategy. He also pointed out that buildings equipped with a CAV system as well as a building with 24-hr occupancy are not good candidates for this type of control strategy.

3.2.1. HVAC systems distribution across office building stock

As it can be seen from the previous section, various HVAC systems have different energy requirements depending on many parameters. However, the main subject of this research is analysis of the most common HVAC systems in the non-domestic building stock, particularly in office buildings. Knight and Dunn (2005) noticed that in the UK there is little information available about the most appropriate systems from the energy efficiency and carbon emissions point of view. Rickaby and Gorgolewski (2000) did an analysis on the occurrences of different building servicing systems in a non-domestic stock in the UK. Analysis was based on data recorded by Sheffield Hallam University (SHU) during energy surveys in the “four towns” study. They categorised building servicing systems into four primary categories: small-scale heating only systems, central plant heating only systems, packaged air conditioning systems which provide mainly cooling, and large-scale HVAC systems. Although the further subclassification of large-scale HVAC systems into several categories such as constant air volume systems, variable air volume systems, fan-coil systems, etc. was proposed, it was not included in the analysis due to small sample and inability to distinguish subcategories in all of the surveyed data.

UK air conditioning market reviews (BSRIA, 2008, BSRIA, 2009) summarised the most significant facts about current trends in the UK HVAC sector. Reports gave a vague picture about today‟s most popular HVAC systems installed in both new and refurbished buildings. It is clear that packaged roof-top systems have become more accepted largely due to investment from the retail sector such as cinemas and fast-food restaurants. Traditionally popular fan-coil based systems have started losing market share mainly to variable refrigerant systems and chilled ceiling systems. The number of sold fan-coil units dropped from 120,000 units in 2002 to slightly above 60,000 units in 2006. The total market size for air handling units has not been changed a lot, but there was a drop in selling the variable air volume terminal units in favour of air-water systems such as chilled ceilings and chilled beams, which resulted in the significant increment in sales of air handling units with an integrated heat recovery unit.

The U.S. Department of Energy published the Building Energy Data Book (U.S.DOE, 2009) where they listed system characteristics for the typical office buildings in the USA based on various surveys, studies, engineering estimates, or engineering judgment. According to them, the typical large office building, which is defined as a building with a floor area higher than or equal to roughly 2,320 m^2 (25,000 ft^2), is most likely equipped with one of the following types of all-air HVAC systems: constant air volume system with reheat or variable air volume system with economiser. On the other hand, the typical small office building relies on packaged single-zone system with or without the economiser. Stocki et al. (2007) in their proposed standardised whole-building simulation assumptions for energy analysis of commercial buildings suggested similar systems to be used in office buildings; variable air volume system in large buildings and packaged unitary system in small buildings. In both cases, systems should be equipped with an economiser unit. Westphalen and Koszalinski (1999) prepared a report on Energy Consumption Characteristics of Commercial Building HVAC Systems in which presented a breakdown of the U.S commercial buildings conditioned floorspace according to a building type and system type. The pie chart in Figure 3.8 shows the distribution of different HVAC systems across the U.S office building stock. The most common systems are packaged systems which condition 43.4% of office floorspace. Central variable air volume systems are the second most common with 22.7% of total floor area, followed by individual air-conditioners installed in 12.3% of office spaces. Central constant air volume systems are used in 11.4% of offices while the least common is central system with fan-coils with only 4.7%. 5.5% of office building space is not air-conditioned at all.

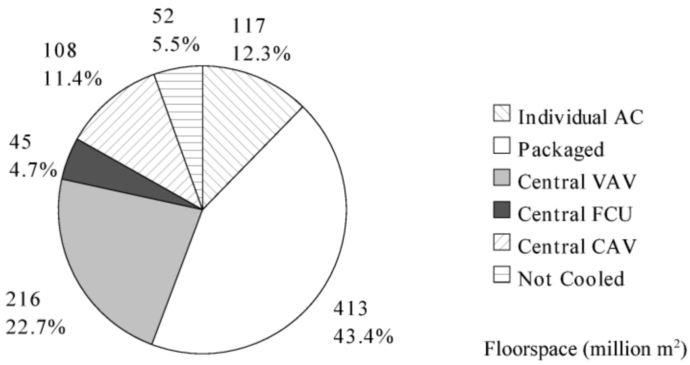


Figure 3.8. U.S. office building conditioned floorspace – system type

3.3. HVAC systems simulation models

It can be concluded from above mentioned that there are two groups of HVAC systems which dominate across office building stock. First group, which include packaged systems and individual air-conditioning systems, is mainly used either in small-sized premises or in individual offices. Central systems, which can be classified into second group, are usually used in larger premises. The most common central systems are variable air volume system, constant air volume system, both equipped with economiser unit, and system with fan-coils. In addition to these systems, it seems that there is a trend in the UK of installing chilled ceilings since recently, according to previously mentioned BSRIA reports. All of this was the primary reason why the following systems were selected for analysis in this research:

- System 1: Variable air volume system (VAV)
- System 2: Constant air volume system (CAV)
- System 3: Fan-coil system (FC)
- System 4: Chilled ceiling system (ChCeil)

Although, each of these systems will be described in more detail in this chapter, some of their basic characteristics are as follows:

- Variable air volume and constant air volume systems models are equipped with an air-side economiser unit in order to reduce energy consumption.
- Air side in both fan-coil system and chilled ceiling system models operates with a constant 100% fresh air flow rate (dedicated air) sufficient only to satisfy fresh air requirements.
- Systems with dedicated air are modelled with a heat recovery unit as an energy saving measure.
- Hot water radiators are used to cover heating demands in chilled ceiling systems.
- System 4, the chilled ceiling system, actually represents two systems, 4a and 4b. System 4a using embedded pipes into concrete ceiling to provide cooling, while system 4b is composed of exposed aluminium panels.

In addition to these systems, which can be considered as realistic representative systems, the system that calculates building zone demands only is also included in the analysis. It has been named System 0 because it is used as a reference system for further comparisons.

3.3.1. System 0: Building demands - Ideal loads air system

Building demand calculations are used in studies where the main task is to investigate the performance of a building and its components. Moreover, comparing building demands is essential for exploring different design options and analysing the influence of various parameters, individually or in combination, on building thermal behaviour. Building demands are usually calculated by taking into consideration only typical heat gains and heat losses that occur in buildings, which are:

- Transmission heat gains/losses through building envelope elements,
- Solar heat gains through fenestration areas,
- Internal heat gains from occupants, artificial lighting and office electrical equipment,

- Infiltration air heat gains/losses, and
- Fresh air ventilation heat gains/losses.

The Ideal Loads Air System model used in the EnergyPlus building simulation software is natural choice for studies where the main task is to evaluate performance of a building with an ideal system. Ideal system supplies air to the zone in sufficient quantity and at the specified condition to meet the zone heating or cooling requirements. The system consumes no energy and it is used for demand calculations. This system can be explained as an ideal unit which mixes air at the zone leaving condition with the specified amount of outdoor air followed by adding or removing heat and moisture at 100% efficiency in order to produce a supply air stream at specified conditions. The volume flow rate of the supply air stream is varied between zero and the maximum depending on the zone loads, which is behaviour quite similar to the variable air volume terminal unit.

3.3.2. System 1: Variable air volume system (VAV)

As mentioned previously, the VAV system varies its supply air volume rate, while keeping a supply air temperature constant, to match the reduction of space load during part-load, to maintain a predetermined space parameter, usually air temperature, and to conserve fan power at reduced volume flow. The model of VAV system with zone reheaters is presented in Figure 3.9. The system is composed of three loops: two water loops (hot and cold water) and one air loop.

3.3.2.1. Water loops in VAV systems

The chilled water loop (blue line in Figure 3.9) connects the chilled water source (CW Source) and the cooling coil (CC) located in the air handling unit (AHU). Hot water loop (red line in Figure 3.9) distributes the hot water from the hot water source (HW Source) to both the heating coil (HC) in the AHU and zone reheating coils installed in zone air terminal units (ATU). Both loops are composed from two parts: primary loop and secondary loop. Primary loops (top left in Figure 3.9) connect heat/cold energy generating equipment with the secondary loops. Hot water and chilled water pumps are installed in primary loops. Secondary loops transfer water from primary loops to equipment in zones and AHU.

Pumps drive hot and chilled water flows around loops. Pumps in HVAC systems can be either constant speed or variable speed. According to the General Information Report GIR 40 (BRESCU, 1996) a constant flow in heating primary loop should be maintained at all times. CIBSE Guide H (CIBSE, 2009) discussed the operation of chilled water primary loops and concluded that the practice in the past was to have a constant flow, while the modern chillers can operate in both regimes: constant flow and variable flow. In this particular model, both hot water and chilled water pumps were selected to operate with a constant speed, which means that they maintained constant flow rates. Flow rates in secondary loops are driven by energy requirements of attached equipment and they are modulated in response to changes in requirements. During partial load operations, when equipment flow rates are reduced, excess hot/chilled water supply is diverted through bypasses. If there are no requirements at all, pumps operation is terminated to preserve energy. This is so-called intermittent pump control. To determine pump energy consumption, it is required to specify a system pressure drop and a pump motor efficiency. System pressure drop depends on a pipe network complexity. As in real buildings the complexity of the pipe network will vary from building to building, for the purpose of this research the EnergyPlus default value of 180 kPa was selected in both hot and chilled water loops. Pump motor efficiency was set to 90% which can be assumed a typical good practice (Stocki et al., 2007).

Another important parameter which characterises water loops is the water temperature regime. Low pressure hot water systems, which are the most common in HVAC systems, operate with a maximum supply water temperature of 82°C and a typical system temperature drop of 10-11°C (CIBSE, 2005a). On the other hand, a chilled water systems operates with a design supply temperature between 4°C and 13°C, usually 7°C, and a temperature rise of 5-6°C (CIBSE, 2009, CIBSE, 2005a, ASHRAE, 2008). As the main aim of this research is to investigate the complexity of buildings and secondary HVAC systems, it was assumed that the primary system operates with 100% efficiency and it is capable of providing enough energy at desired temperature to fulfil all requirements all the time. Supply water temperatures in models in this research were set to 82°C in the hot water loop and 7°C in the chilled water loop, with a temperature difference of 11°C and 5°C respectively.

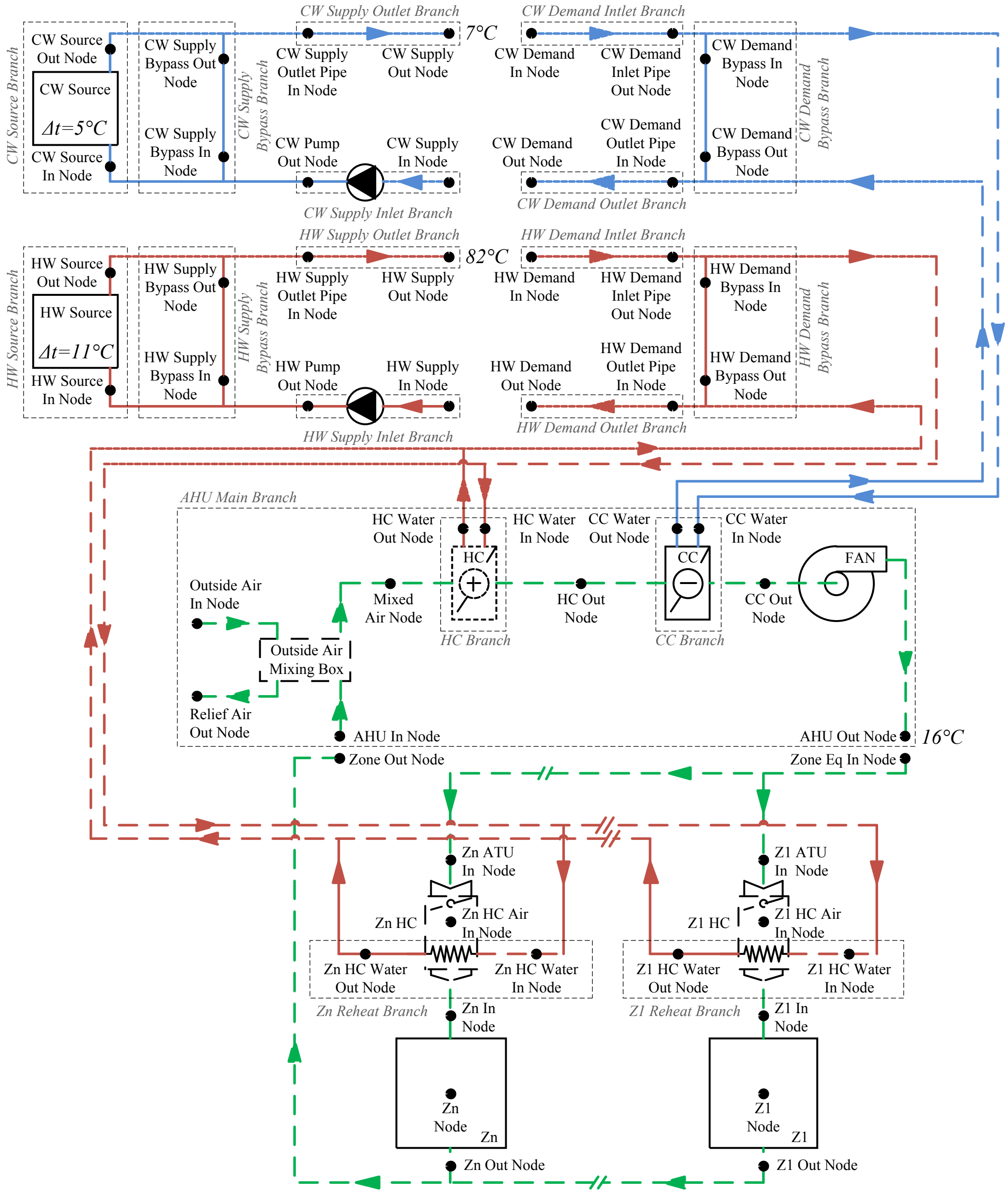


Figure 3.9. Variable air volume system (VAV)

3.3.2.2. Air loop

Air loop (green dashed line in Figure 3.9) delivers the conditioned air into zones in order to respond to zone heating/cooling demands and fresh air requirements. Outdoor air is mixed with the return air stream in the outside air mixing box. Mixed supply air is passed over heating and cooling coils where it is treated, if there is a need, to meet the supply air temperature setpoint (AHU Out Node in Figure 3.9). Supply air is delivered to individual zones through zone air terminal units where it is additionally reheated according to zone requirements.

The supply air temperature setpoint is an important parameter to be considered during design process. The optimum value can be determined, however it is case based because it is affected by many parameters such as: climate region, operating hours, internal heat gains, free cooling availability, efficiency of HVAC equipment, etc. (Ke and Mumma, 1997, Engdahl and Johansson, 2004). The minimum supply air temperature has to be higher than the air dew point temperature in conditioned zones in order to avoid condensation. In addition, the risk of feeling drafts has to be taken in consideration, as it is higher when this temperature is lower. The minimum supply air temperature is usually limited to 12°C (CIBSE, 2009). For a comfort applications, the usual range of temperature differences between desired zone air temperature and supply air temperature during cooling operation is between 8°C and 12°C (CIBSE, 2005a). The high-end values are recommended for high ceiling applications. If the mixing between supply air stream and indoor air cannot be guaranteed, the temperature difference should not be higher than 5°C. In this particular model, the supply air temperature was limited to 16°C, which allows temperature difference of 8°C as the air temperature setpoint in office zones was set to 24°C.

The supply air temperature setpoint was also used to control a water flow rate through both heating coil and cooling coil in the air handling unit. The water flow rate through the coil is increased when more heating or cooling is requested, which results in increased heat exchange between the water and the supply air stream. Preconditioned air is then split and delivered to the zones through the air terminal units where, if there is a need, is additionally heated. Each air terminal unit is composed of a damper and hot water coil, both operated by zone temperature sensor, with a reverse damper action. This means that in the heating mode the unit starts at minimum air flow and minimum hot water flow. When the heat load increases, the hot water flow is increased until it reaches maximum flow. If the load still cannot be met, the air damper starts to open. In the cooling mode, the damper modulates between the minimum and maximum air flow rates in order to maintain the room setpoint. The ratio between minimum and maximum air flow rates is called turndown ratio. It is usually specified to meet minimum air quality requirements. Turndown ratio is typically between 0.3 and 0.5 (Cho and Liu, 2009a). Stocki et al. (2007) recommended the turndown ratio of 0.3 to be used in a VAV system simulations which was the value used in this research.

Securing variable flow rates in VAV system applications can be obtained by changing either fan characteristics or system resistance. Fan characteristics can be changed by using fan inlet dampers or by varying a fan motor speed, while a system performance changes by changing a system resistance, usually by modulating discharge dampers. Fan power consumption at part load operation is calculated in EnergyPlus by using a simple fan model presented in HVAC Secondary Toolkit (Brandemuehl, 1993). Firstly, the fraction of full load power (FFLP) is determined as a cubic function of the part load ratio (PLR), where the PLR is ratio of the actual air flow rate to the designed flow rate. The actual fan shaft power is calculated by multiplying the fraction of full power with the fan full load power. The fan full load power depends on the design air flow rate, fan total pressure rise and total fan efficiency. Fan total electricity consumption is calculated by dividing the fan full load power with the fan motor efficiency. Typical value for the fan motor efficiency is around 0.9. On the other hand, the total fan efficiency depends on a fan type and ranges between 0.6 and 0.85 for the most common fan types used in HVAC applications (Wang, 2000). Low-end values are typical for forward-curved centrifugal fans, while high-end values are usual for backward-curved centrifugal fans. Value of 0.7 was selected for the total fan efficiency as both backward-curved and forward-curved centrifugal fans can operate with such efficiency (ASHRAE, 2008).

Another parameter that has to be defined to perform simulations in EnergyPlus is the fan total pressure rise. Ducts and fittings losses, as well as resistance of HVAC components such as dampers, coils, heat recovery units, etc. affect the fan total pressure rise requirements. CIBSE Guide B (CIBSE, 2005a) presented pressure drops in HVAC applications. Typical values range between 0.5 kPa and 1 kPa for low and medium velocity systems. The EU Standard EN 13779:2007 (CEN, 2007a) listed pressure drops for specific components in air handling systems. Total pressure drop, calculated by summarising the most common components pressure drops, is between 0.6 kPa and 1.1 kPa in applications with low and normal pressure drop.

CIBSE Guide F (CIBSE, 2004) however uses Specific Fan Power (SFP) as a parameter which quantifies an energy efficiency of fan powered systems. SFP is defined as a ratio of the fan total electricity power to the fan design air volume flow rate. The SFP values between 1.5 kW/(m^3/s) and 3 kW/(m^3/s) are typical practice in the UK office buildings. To achieve good practice, the SFP should be less than 2 kW/(m^3/s), although very efficient systems can sometimes achieve around 1 kW/(m^3/s) (CIBSE, 2004). Current building regulations limit the maximum specific fan power in air distribution systems. The UK Non-domestic Building Services Compliance Guide (DCLG, 2010) listed the maximum SFP"s in new buildings as well as in existing buildings. Central mechanical ventilation system including both heating and cooling has to have specific fan power less than 1.8 kW/(m^3/s) in new buildings or 2.2 kW/(m^3/s) in existing buildings. These values can be extended for additional components such as heat recovery unit (0.3 kW/(m^3/s)) or additional filter in return air stream (0.1 kW/(m^3/s)), etc. American standard 90.1-2007 (ANSI/ASHRAE, 2007b) limits the specific fan power to 2.3 kW/(m^3/s) in variable air volume systems, while the limit is even more strict in constant air volume applications and amounts 1.7 kW/(m3/s).

Taking into consideration specific fan power limits and typical values for the total fan efficiency and the fan motor efficiency of 0.7 and 0.9 respectively, the fan total pressure rise should not exceed 1.1 kPa. In this research, the fan total pressure rise was set to 0.9 kPa, which can be assumed good practice in low velocity HVAC applications.

3.3.2.3. All-air systems energy saving measure - air-side economiser

The design air flow rate in all-air systems is in most cases higher than fresh air requirements because it is determined according to cooling demands. Many air handling systems employ mixing box where the portion of a return air from occupied areas is recirculated and mixed with an intake of outside air. Mixing box employs three sets of dampers, controlled either by temperature or enthalpy sensors, which can vary the proportion of outside air to provide economic operation by maximising the benefits from free cooling. A box equipped with this type of control is often called an air-side economiser and it has become popular energy conservation measure. ASHRAE Standard 90.1-2007 (ANSI/ASHRAE, 2007b) even made a requirement of the usage of an air-side economiser in systems which condition areas are larger than 50 m^2.

Yao and Wang (2010) studied the influence of both types of air-side economisers, temperature controlled and enthalpy controlled, on annual energy consumption of the VAV system in office building. They also investigated the impact of climate by using six typical climate regions in China. General conslusion was that the performance of the VAV system can be significantly improved by implementing an air-side economiser. The total energy reduction was between 10% and 20% in hot and humid climates, while in cold and dry regions it was around 5-10%. In addition, their recommendation was to implement enthalpy control in hot and humid regions and temperature control in cold and dry climates. Korolija et al. (2009) analysed the impact of an air-side economiser on CO_2 emissions in the UK office building with two levels of insulation, low-level and insulated according to the current best practice. The reduction in CO_2 emissions in buildings equipped with a VAV system was 7.3% for low-level insulation and 12.6% in case of best practice insulation. In building with a CAV system, these percentages were even higher: 9.5% and 15%.

Both all-air HVAC systems, VAV and CAV, developed for the purpose of this research are equipped with a temperature controlled air-side economiser. Economiser control (Figure 3.10) can be easily implemented to a conventional air handling unit control. The economiser controls the mixed air temperature ($t_{a,m}$) entering coils to a desired setpoint by mixing outdoor air with return air. This setpoint is below the supply air temperature setpoint ($t_{a,s}$) as it takes into account the downstream system fan heat which can increase the air temperature up to 2°C (Oughton and Hodkinson, 2008).

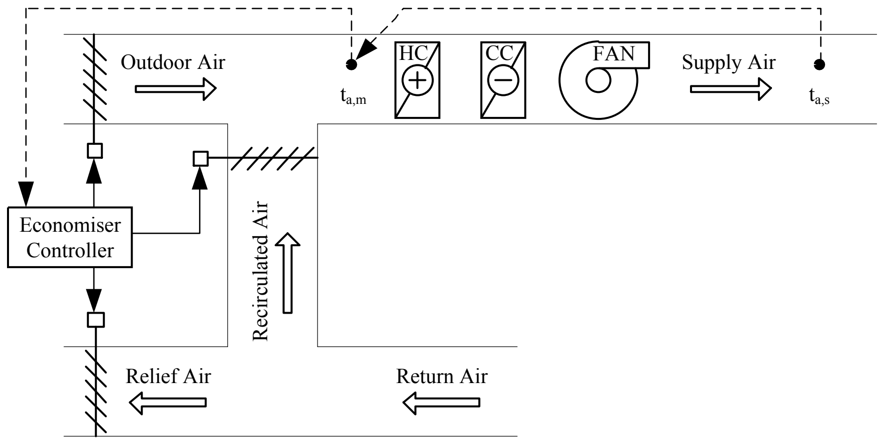


Figure 3.10. Economiser control scheme

The air-side economiser functioning can be divided into four operating regimes (Figure 3.11). First operating regime is when the outdoor temperatures are very low. Due to that, the mixed air temperature setpoint can only be achieved with significantly decreased outdoor airflow rate. If the outdoor airflow rate is lower than the fresh air requirements, the airflow rate is set to the minimum required for ventilation, which leaves the resultant mixed air temperature lower than the setpoint. Therefore, the heating coil will operate to achieve supply air temperature setpoint.

Second operating regime starts when the desired mixed air temperature setpoint is achieved by mixing two air streams while the requirement for minimum ventilation is satisfied. It ends when the outdoor temperature becomes equal to the desired setpoint, which makes system operating with 100% fresh air. During second operating regime, there is no need either for cooling or for heating.

Third operating regime lasts until both the return air temperature and the outdoor air temperature are above the mixed air temperature setpoint. During that time, the supply air temperature setpoint cannot be maintained without using cooling coil. However, as the outdoor temperature is below the return air temperature, it is desirable to operate with 100% fresh air to reduce cooling requirements.

Last operating regime starts when there are no energy benefits provided by economiser. At that point, the outdoor air flow rate is reduced to minimum required for ventilation.

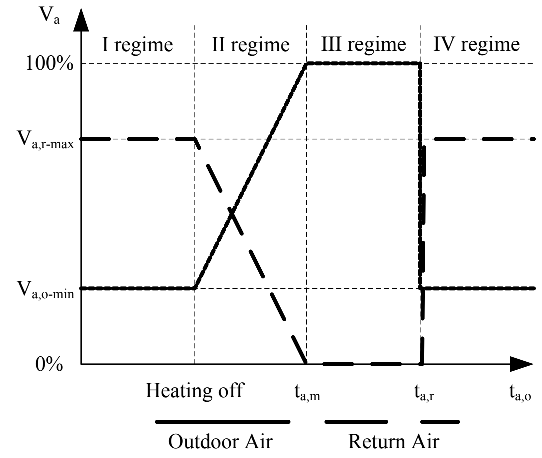


Figure 3.11. Economiser control regimes

3.3.2.4. VAV out-of-hours system operation

During occupied hours (weekdays between 7am and 7pm) the system is running all the time with no restrictions. On the other hand, during unoccupied hours, it is, by default, switched off completely. However, if the air temperature in any zone drops below heating setback setpoint or exceed cooling setback setpoint, the system will start operating in so-called cycling mode. Cycling mode operation is limited by setback temperature offset, which was set to 1°C. This means that the system will start working when the air temperature in any zone drops/rises by 1°C below/over heating/cooling setback setpoint. It will continue working until the air temperature in the zone which "called" for conditioning exceeds 2°C increment/decrement, depending on a heating/cooling operation. In addition, during cooling cycling period, the chilled water pump is kept switched off whenever is possible to offset the demand with free cooling.

3.3.3. System 2: Constant air volume system (CAV)

EnergyPlus model of the constant air volume system is presented in Figure 3.12. It is very similar to the previously described model of the variable air volume system with three major differences. First difference is related to the supply fan type. In contrast to the VAV system, in CAV applications a constant speed fan distributes same, designed air flow rate all the time, which results in continuous high power consumption. Operating with a constant air flow rate requires less complex air terminal units, which is the second difference. Namely, air terminal units do not have modulating dumpers.

Third difference is the supply air temperature setpoint (AHU Out Node in Figure 3.12). The VAV system operates with a constant supply air temperature setpoint, while in the CAV system this value is variable, which is one of system basic characteristics. The supply air temperature in the CAV system is adjusted according to the cooling demand of the warmest zone, which means that the setpoint is always the highest possible temperature which can meet the cooling requirements of all zones. This is, however, subject to minimum and maximum setpoint constraints. This strategy reduces reheating energy consumption when compared to operating with a fixed supply air temperature setpoint. Minimum and maximum supply air temperature was set to 16°C and 22°C respectively.

All other aspects of the system operation, such as chilled/hot water temperature regimes, air-side economiser, out-of-hours operation, etc. are completely the same as in the VAV system.

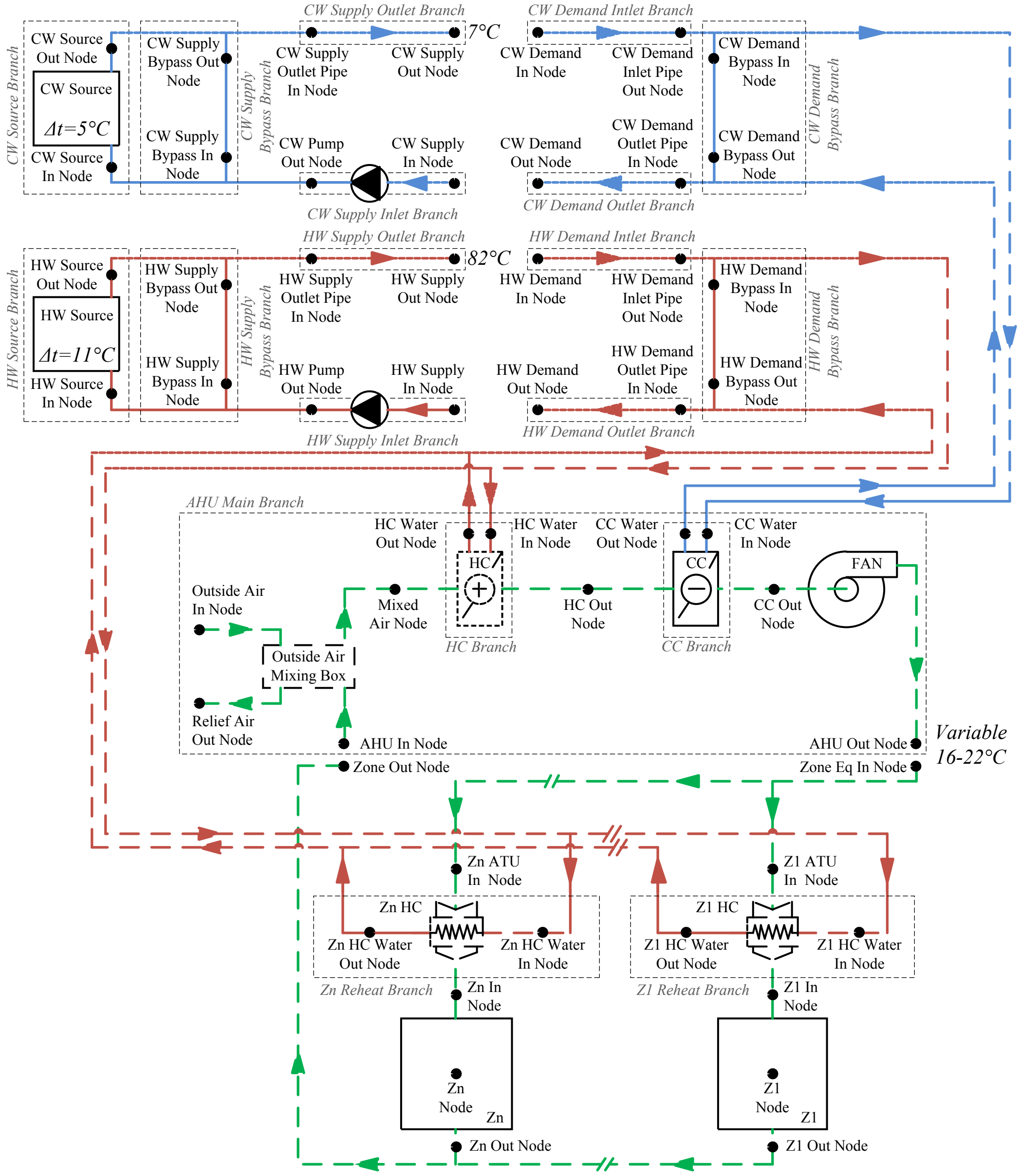


Figure 3.12. Constant air volume system (CAV)

3.3.4. System 3: Fan-coil system (FC)

The diagram of the HVAC system utilizing fan-coils is presented in Figure 3.13. A fan-coil unit is a packaged assembly composed of coils, air circulating fan and filter. There are two types of fan-coil units: two-pipe and four-pipe. Two-pipe unit employs single coil which is used for cooling only (non-changeover) or for both cooling and heating (changeover). In changeover applications, chilled or hot water is distributed from the sources to units via three-port changeover valves. This type of fan-coil systems is appropriate only in climate regions with apparent summer/winter seasons, which is not the case in the UK (CIBSE, 2005a). For the UK climate, the most suitable is a system with four-pipe fan-coil units, which integrate separate heating and cooling coils, and this particular model is based on such units.

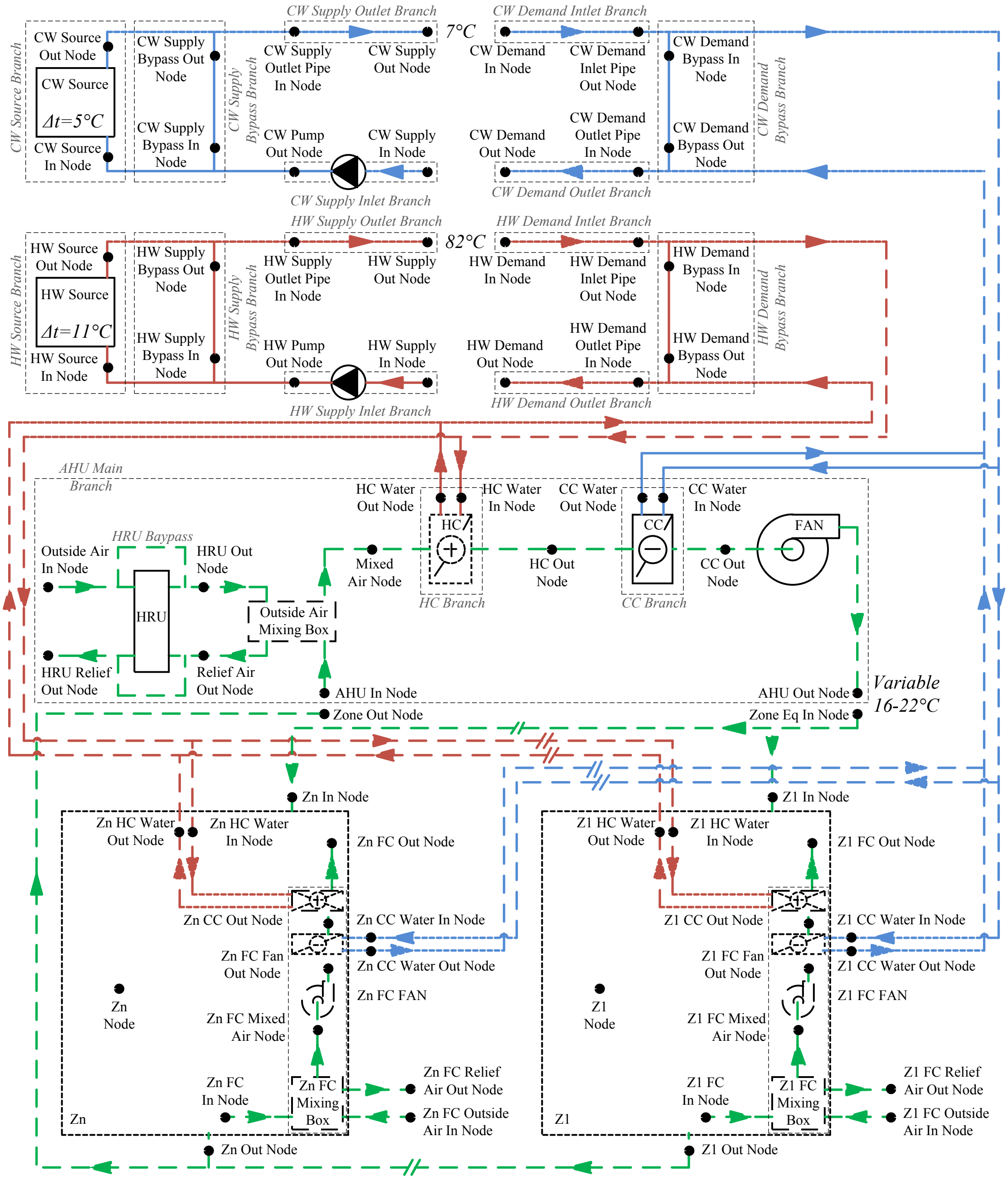


Figure 3.13. Fan-coil system (FC)[1,2]

[1] The „Outside Air Mixing Box" element in the main AHU is presented in this Figure since it is an EnergyPlus required input. In this particular system, it was set to operate with 100% fresh air all the time.
[2] The Mixing box element in the zone fan coil unit has an ability to operate with outdoor fresh air, although in this study the amount of outdoor air delivered to the zone through this element was set to 0.

Fan-coil units control space temperature by recirculating room air, although in some applications a portion of outdoor air can be drawn through an outside wall. The unit capacity can be controlled in three ways: by varying water flow rate through coils, by changing a fan speed, or by air bypass system. Waterside control was used in the model. During periods with no heating/cooling demand, the fan-coil operation was terminated.

Fans used in fan-coil units are commonly less efficient when compared to fans used in conventional air handling units. Typical total fan-coil fan efficiency is around 50%, while the central air systems have the fan efficiency around 70%. However, the advantage of a fan-coil unit is the maximum allowed specific fan power (SFP), which is significantly lower than typical SFP‟s in all-air systems. While the good practice in all-air systems in UK office buildings is to have the SFP up to 2 kW/(m^3/s), the latest building regulations limits the maximum specific fan power in fan-coil units to 0.6 kW/(m^3/s) (DCLG, 2010). Most manufacturers claim that their fan-coil units operate with a SFP values between 0.3 and 0.4 kW/(m^3/s), although the state of the art equipment can work with SFP‟s as low as 0.15 kW/(m^3/s). SFP value and fan efficiencies are important as they are used to determine fan pressure rise which is one of inputs required by the fan-coil model in EnergyPlus. To achieve performance of the most efficient fan-coil units, the fan pressure rise should be around 75 Pa. On the other hand, it should not exceed 300 Pa due to building regulation limitations. Typical fan-coil units have the fan pressure rise between 150 and 200 Pa. In this research, this value was set to 150 Pa.

Ventilation in fan-coil systems is usually provided by a separate central air system. The central air system supplies fresh air only which results in decreased fan energy requirements than requirements of all-air systems. The ventilation air was supplied at constant flow rate but with a variable temperature which was varied between 16°C and 22°C. Neutral temperature (22°C) minimises ventilation loads on the fan-coils during heating period, while supply air temperature was allowed to decrease in order to maximise benefits from free cooling during warmer weather.

The central air handling unit and accompanied distribution ductwork were sized according to fresh air requirements and they were significantly smaller than AHU and duct network in all-air systems. Despite much smaller air flow rate delivered to zones, when compared to all-air systems, the air pressure drop across the central air system in fan-coil application can be assumed to be the same (0.9 kPa). The absence of mixing box equipped with economiser, zone air terminal units and reheating coils which are common in all-air systems, can be compensated with a heat recovery unit which can create additional pressure drop as high as 370 Pa (ASHRAE, 2008).

Chilled water loop and hot water loop had the same parameters as loops previously described in the VAV system. On the other side, the system operation during unoccupied period differs from that adopted in all air systems. Any demand, either cooling or heating, during unoccupied period was covered by in-zone fan coil units. The central air system was kept off during that time.

3.3.4.1. Dedicated air systems energy saving measure – heat recovery unit

Heat recovery unit (HRU) was used in this particular model because it is a very popular energy saving measure in systems which work with 100% fresh air. Heat recovery devices in ventilation systems recover energy from the exhaust air stream. They are beneficial during both heating and cooling periods, although in moderate climate such as is the UK they have negligible influence on system cooling demand. The reason for this is that the outdoor air temperature is higher than return air temperature for a very limited time during summer season. In particular, for the London Gatwick weather file, the total number of hours when the outdoor temperature is above 24°C, which was cooling setpoint, is 118 of which 39 hours occur during weekend when the system in office building is usually switched off.

Three most popular heat recovery devices are: thermal wheel, air to air plate heat exchanger and run around coil (CIBSE, 2009). For the purpose of this study it was decided to equip air system with a HRU, in particular with a plate heat exchanger, due to this measure can reduce office building total CO_2 emission up to 10% when compared to the system without HRU (Korolija et al., 2011). Typical air to air plate heat exchanger effectiveness is between 50% and 80% (CIBSE, 2005a, ASHRAE, 2008), where the effectiveness is defined as the ratio of actual heat transfer to maximum possible heat transfer. Non-domestic building services compliance guide recommended that the minimum sensible heat recovery effectiveness for plate heat exchangers in new and existing buildings should not be less than 50% (DCLG, 2010). American Air conditioning, Heating and Refrigeration Institute (AHRI) provides a list of certified plate heat exchangers according to their standards of testing HVAC equipment where can be seen that the maximum certified plate heat exchanged has the sensible heat recovery effectiveness 83% (AHRI, 2011). Average effectiveness of certified units, which have effectiveness‟s above 50%, is 65%, which was the value of HRU sensible effectiveness used in this study. The HRU in the model was also equipped with bypass dampers to maximise free cooling by bypassing the heat exchanger.

3.3.5. System 4: Chilled ceiling system (ChCeil)

The last developed model was the HVAC system with the chilled ceiling (Figure 3.14). This system shares some elements with the fan-coil system. Characteristics of the air loop and the operation of the ventilation system are exactly the same. Moreover, the hot water loop parameters, which include hot water temperature regime, constant speed circulation pump and system pressure drop, are identical as in the fan-coil system. Chilled water pump and pressure drop in the chilled water loop are also identical. The major difference in the chilled water loop is the chilled water temperature regime which will be explained later in the chapter.

Where this particular system differs from the fan-coil model described previously is in the type of an HVAC equipment used to cover heating and cooling demands. Heating demand is covered by radiator heating system. A zone thermostat controls hot water radiators located in each zone by modulating hot water flow rate in response to the zone heating requirements. Cooling requirements are covered by a radiant cooling system, in particular chilled ceiling system. Radiant cooling systems provide cooling by a combination of convection and radiation (between 50% and 60% of the heat is transferred by radiation) in contrast to conventional HVAC systems which deliver cooling almost completely by convection only.

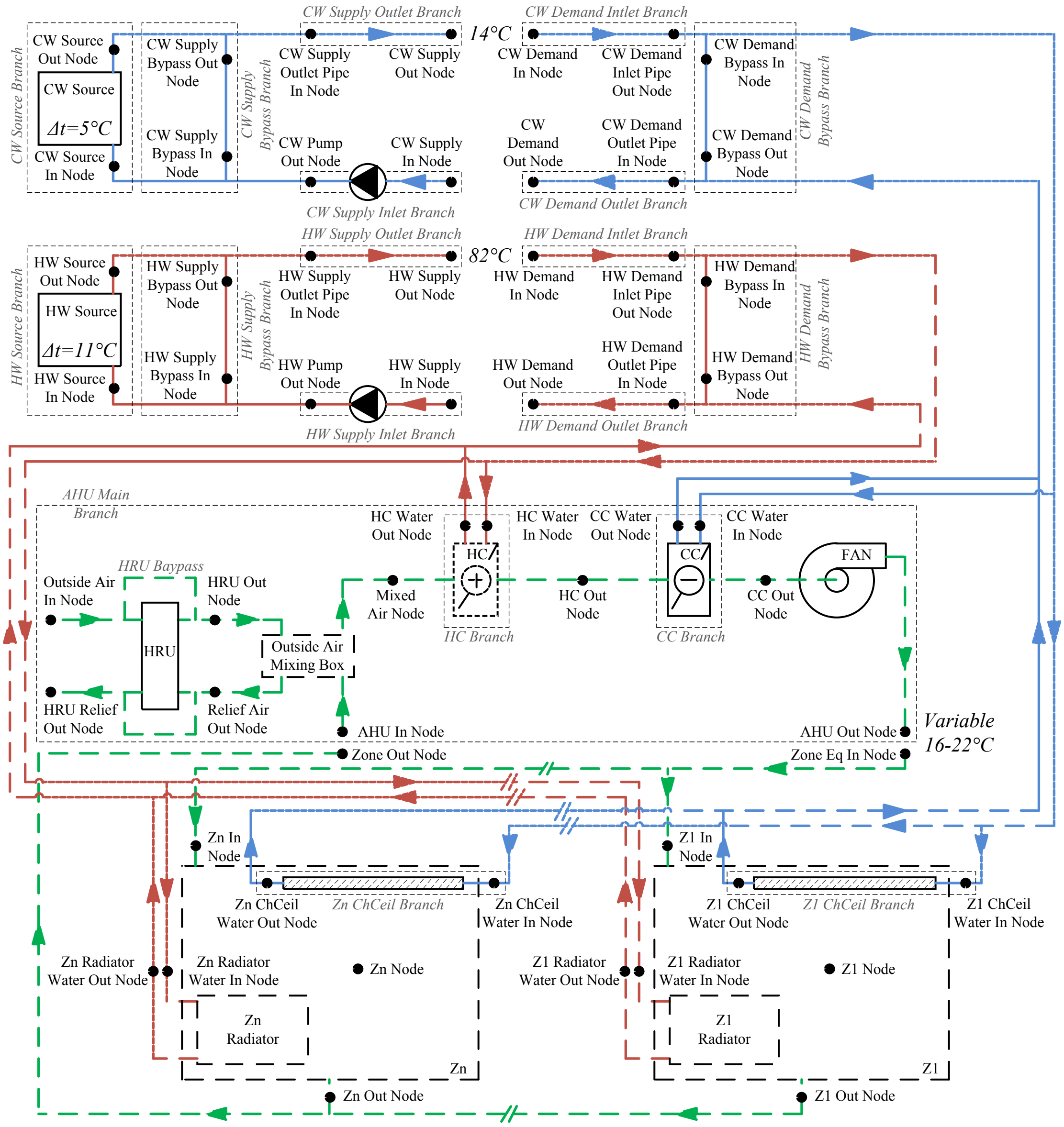


Figure 3.14. Chilled ceiling system (ChCeil)[3]

Chilled ceilings can be classified into two categories: thermally lightweight systems and heavyweight applications. Thermally lightweight systems usually employ a panel type construction (Figure 3.15) and they are often called radiant panel systems. Radiant panels are made of metal sheets (usually aluminium) to which the chilled water pipes are welded. On the other side, the thermally heavyweight system uses chilled water pipes directly embedded in a ceiling (Figure 3.16), sometimes called activated ceiling. The characteristic of this system type is to have high thermal capacity to store

[3] The „Outside Air Mixing Box" element in the main AHU is presented in this Figure since it is an EnergyPlus required input. In this particular system, it was set to operate with 100% fresh air all the time.

energy which results in a slow response to load changes. Both chilled ceiling system types, radiant panels and activated ceiling, are often made with an insulation layer placed behind pipes to prevent back heat losses. In this research, models of both systems were created and simulation results were analysed.

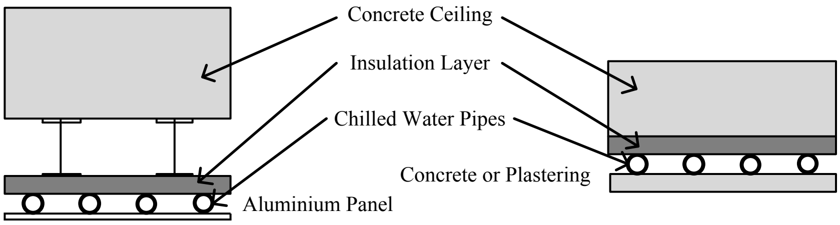


Figure 3.15. Radiant panels **Figure 3.16. Embedded pipes**

Chilled water pipes, either embedded into concrete ceiling or attached to radiant panels, affect the ceiling surface temperature, which result in cooling energy distributed partially by convection heat transfer between surface and room air and partially by radiation. According to Mumma (2002) and Novoselac and Srebric (2002), as a result of these two occurrences, the room dry-bulb air temperature with the chilled ceiling system can be approximately 2°C higher to obtain the same thermal comfort as with the conventional all-air system. Due to that, the cooling temperature setpoint was increased from 24°C to 26°C in offices and from 26°C to 28°C in common areas. Positive side effects of temperature setpoint shift are that the building conduction losses are decreased as well as ventilation losses during cooling period. In addition, the ventilation air can remove more heat since there is a 2°C higher temperature difference between indoor air and supply air.

Despite the fact that chilled ceilings are to some extent self-regulating, since the cooling output increases with the room temperature rise, the control of the cooling circuit is quite similar to one described in the fan-coil model. Chilled water flow rate is modulated in response to a zone temperature sensor from switched off periods up to maximum designed flow rate. The major difference in comparison with conventional HVAC systems is the chilled water supply temperature which has to be controlled due to risk of condensation on the surface of chilled panels or ceiling. Condensation will start to occur if the surface temperature drops below the zone air dew point temperature. To avoid this, the chilled water temperature should be between 13°C and 18°C (CIBSE, 2005a). CIBSE Guide H (CIBSE, 2009) stated that the minimum chilled water temperature is typically 14°C which was the value used in this research.

3.4. HVAC system model summary

For the purposes of this research, four types of HVAC system models were developed. The first is the variable air volume system (VAV) and the second is the constant air volume system (CAV). Both systems belong to the all-air HVAC system group and are equipped with zone reheating boxes. The last two systems are air-water systems, in particular Fan-coil units with a dedicated outside air system (FC) and the chilled ceiling system. The chilled ceiling system is also equipped with an air handler which delivers only fresh air while the heating demand is provided by radiator heating.

The VAV System (Figure 3.17a) varies its supply air volume rate while keeping a supply air temperature constant to match the reduction of space load during part-load and so maintain a predetermined zone dry-bulb air temperature while conserving fan power at reduced volume flows. The main heating (HC) and cooling (CC) coils are controlled according to the supply air temperature (t_{sa}) which is set to 16°C. Preconditioned air is delivered to the zones through the air reheating boxes where it, if there is a need, is additionally heated. Each air reheat box is composed of a damper and hot water coil both operated by zone temperature sensor (t_{za}) with a reverse damper action. This means that in the heating mode it starts with minimum air flow and minimum hot water flow. With a load increment the hot water flow is increased until it reaches maximum flow, then the air damper starts to open to meet the load. In contrast to the VAV system the CAV system (Figure 3.17b) keeps the air volume flow rate constant while varying its supply air temperature (t_{sa}) from 16°C to 22°C according to the cooling demand of the warmest zone. This strategy minimizes zone reheat coil energy or overcooling.

The amount of the outdoor air in both systems is controlled via an outdoor air mixing box equipped with an economizer which mixes return air and outdoor air in proportion to meet the mixed air temperature setpoint (t_{ma}). The mixed air temperature is lower than supply air temperature by around one degree centigrade because the supply air stream, by passing over the fan motor, absorbs the fan dissipated heat. By using the economizer unit the amount of outdoor air is increased whenever it is possible to benefit from free cooling.

The fan-coil system shown in Figure 3.17c is composed of zones with four-pipe fan coils and an air handing unit which distributes 100% fresh air, which is enough to meet fresh air requirements only. Fresh air supply temperature is controlled to vary the supply air temperature between 16°C and 22°C in order to maximise the benefits of free cooling. However, free cooling is very limited due to a significantly lower supply air volume flow rate in comparison with the VAV and CAV systems. Each fan-coil unit is composed of a fan which recirculates room air, along with heating and cooling coils. The indoor temperature is controlled according to a local thermostat (t_{za}) which varies the water flow rate through the heating or cooling coil in response to the zone demand. In cases when there is no need for heating or cooling the fan coil fan is switched off. The outdoor air is pre-treated using a heat recovery unit (HRU) with 65% effectiveness and this exchanges heat between the supply air stream and the exhaust air stream.

The chilled ceiling system (Figure 3.17d) is composed of the following elements: a chilled ceiling element; an air handling unit equipped with a heat recovery unit which delivers only fresh air (the air side is controlled in the same way as in the Fan-coil system); and radiators to meet heating demand. Due to the way this system delivers cooling and maintains comfort (partially by radiation and partially by convection), the zone cooling temperature setpoint was increased by 2°C; from 24 to 26°C in offices and from 26 to 28°C in common areas. Two types of chilled ceiling element were investigated in this research: a thermally lightweight element (aluminium panel), and a heavyweight application (chilled water pipes embedded directly into the concrete ceiling).

In all the abovementioned systems it was assumed that the primary HVAC system operates with 100% efficiency and it is capable of providing enough energy at the desired temperature to fulfil all requirements all the time. The hot water temperature delivered from the primary system was set to 82°C in all the systems studied and the chilled water supply temperature was set to 7°C except in the case of the chilled ceiling system where it was set to 14°C.

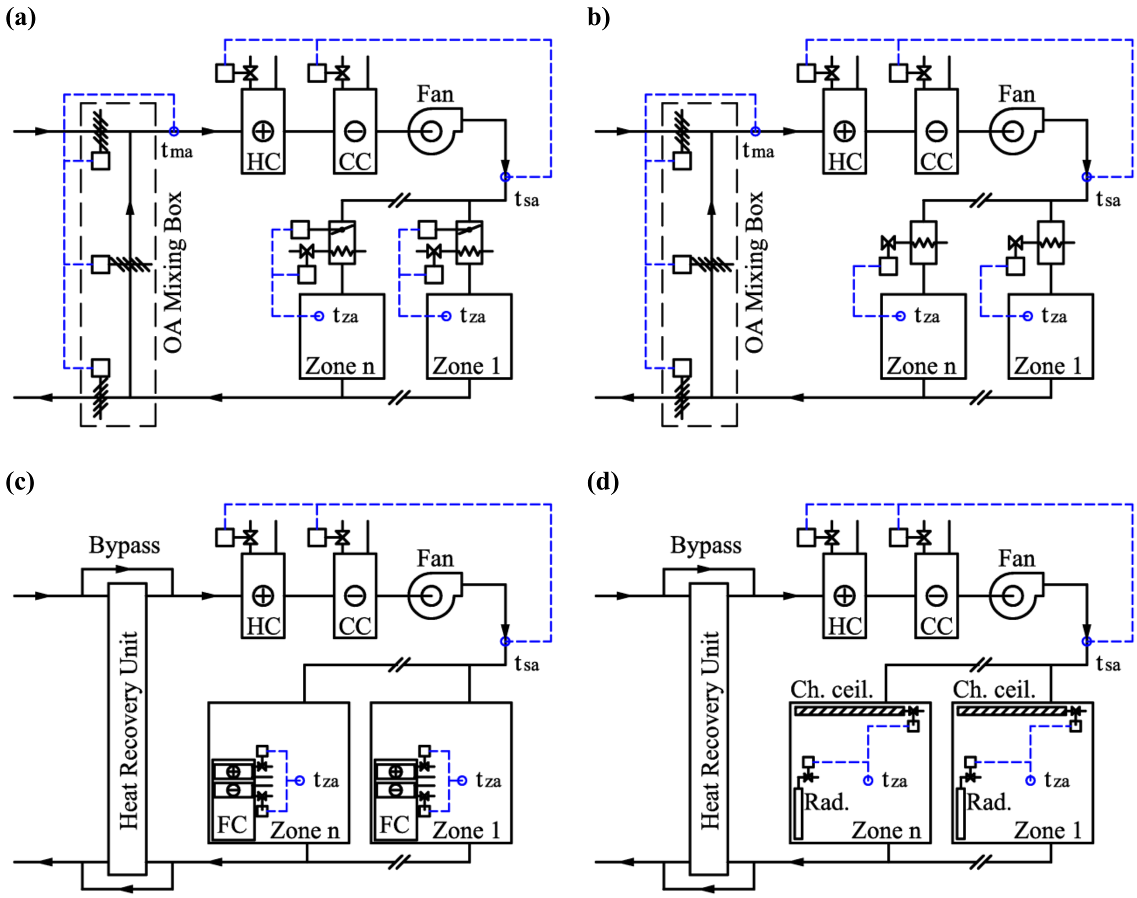


Figure 3.17. HVAC system models: (a) Variable air volume system, (b) Constant air volume system, (c) Fan-coil system, and (d) Chilled ceiling system

Chapter 4

Simulation Results and Analysis

Previous chapters described office building parameters and HVAC simulation models which are used in a large parametric study. The basis for the parametric study were models of four office building types, which represent the most typical office building build forms originally presented in Figure 2.8:

- Type 1 – open-plan sidelit buildings (OD),
- Type 2 – cellular sidelit buildings (CS),
- Type 3 – artificially lit open-plan buildings (OA), and
- Type 4 – composite sidelit cellular around artificially lit open-plan buildings (CDO).

These four models were coupled with five types of building fabrics in order to represent a range of existing buildings constructed during the past decades. Each type correspond to the particular Building Regulations and includes the current UK Best Practice, Part L 2002, Part L 1995, Part L 1990, and buildings with minimum or no insulation at all. Variations in fenestration areas were covered by three levels of glazing ratio; 25%, 50% and 75%. Furthermore, two measures of reducing solar heat gains were considered as possible design option: replacing standard glazing with reflective glazing and adding overhangs above windows. Reducing internal heat gains and artificial lighting electricity consumption were achieved by implementing daylight control. Finally, the orientation of buildings was also investigated by rotating the buildings at 45 degree intervals starting from the north. Combination of these parameters gives 3,840 scenarios, where each scenario represents one particular office building as indicated in Figure 4.1.

These buildings were further coupled with five HVAC systems:

- Variable air volume system (VAV),
- Constant air volume system (CAV),
- Fan-coil system with dedicated air (FC),

- Chilled ceiling system with embedded pipes (EMB), and
- Chilled ceiling system with aluminium panels (ALU).

In order to establish the comparison benchmark, the ideal loads HVAC system was also coupled with all building types and simulated. Figure 4.1 shows all variations which were defined by combining individual building parameters with HVAC system models. The total number of possible scenarios is 23,040.

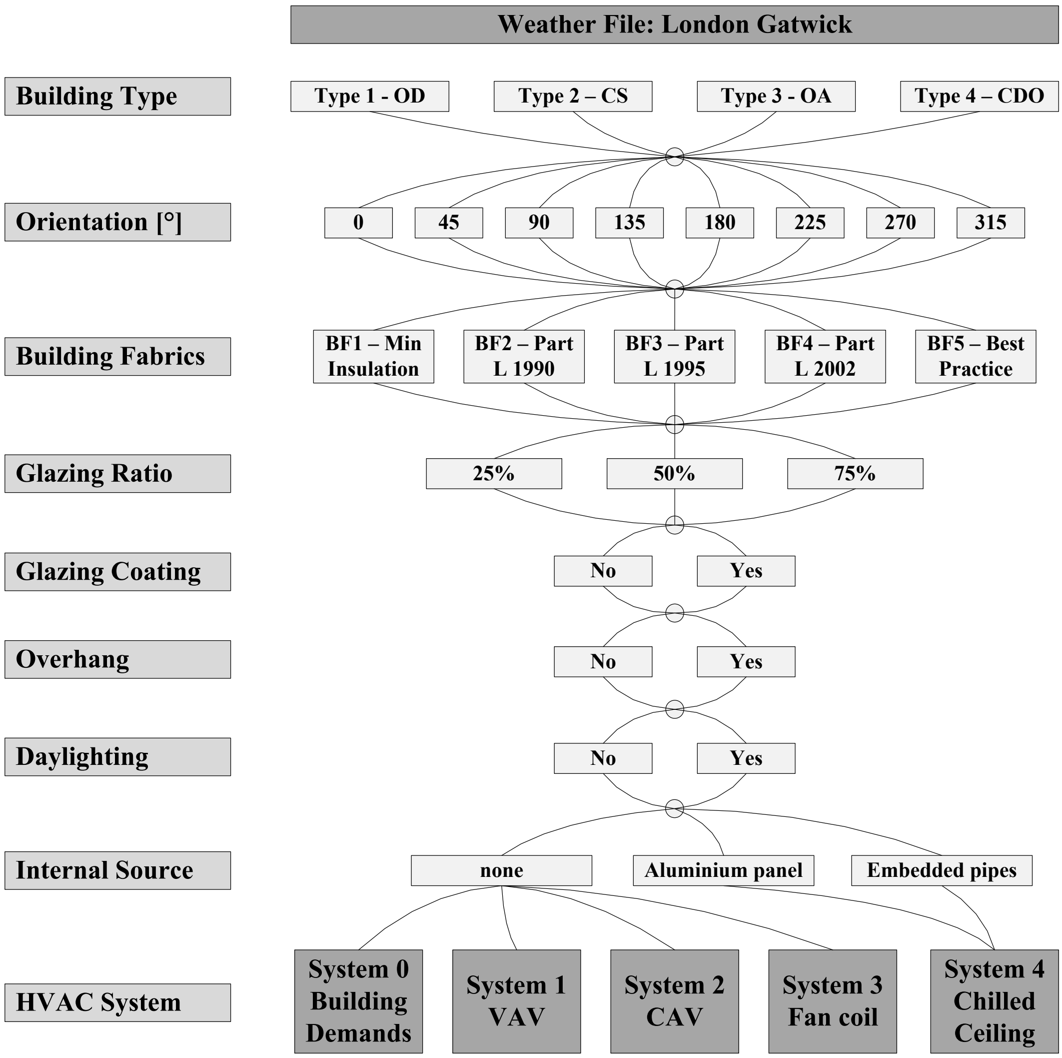


Figure 4.1. Parameter tree[4]

Each of the total 23,040 scenarios represents one building equipped with one HVAC system for which annual simulation was run in order to calculate energy

[4] The „Intemal Source" parameter in Figure 4.1 refers to a type of hydronic equipment installed in a ceiling construction element.

consumption. Simulations were done in the EnergyPlus v.6.0 by using a London-Gatwick weather file (EnergyPlus, 2010j), which is actually the EnergyPlus weather file (EPW) based on the International Weather for Energy Calculations (IWEC) data. This weather file represents a typical meteorological year and consists of hourly data for 12 typical months selected from multi-year data set (more than 20 years). Crawley (1998) recommended using this type of weather data for simulating commercial buildings since it is capable of providing users with energy simulation results that most closely represent typical weather patterns. However, for sizing the HVAC system and equipment, summer and winter design days were used. Design days represent near-extreme weather conditions. London-Gatwick has been chosen as a location for all simulations since it represents the most densely populated area of the UK where most of the commercial activities are located.

The simulations were not set manually, as this would be very difficult, if not impossible. Simulations were run by using jEPlus in conjunction with EP-Macro. jEPlus (Zhang, 2009) is a Java based EnergyPlus shell suitable to manage and run large and complex parametric simulations. It allows users to describe parameters by using graphical interface, and then automatically creates and run EnergyPlus simulations. EP-Macro (EnergyPlus, 2010a) is a part of EnergyPlus package used as a pre-processing tool which provides to users several advanced functions. Working with EP-Macro, jEPlus allows users to prepare large parametric simulation study by using only a limited number of EnergyPlus input files. In this particular research, 49 input files, plus 1 weather file, were required to run 23,040 EnergyPlus simulations. Zhang and Korolija (2010) gave more in-depth explanation of setting and running large parametric EnergyPlus study with a minimal number of input files by using jEPlus and EP-Macro.

During the course of this research, the EnergyPlus (EnergyPlus, 2010b) was chosen as a simulation software due to its powerful capabilities, reputation and free access. Although relatively new, the first version was released in April 2001, EnergyPlus has its roots in two programs developed and released in late 1970s and early 1980s, BLAST and DOE-2 (Crawley et al., 2001b). Like its predecessors, EnergyPlus is an energy analysis and thermal load simulation program. Based on the most popular features and capabilities of BLAST and DOE-2 and extended with many new program modules, it calculates the heating and cooling demands necessary to maintain desired indoor thermal conditions, secondary HVAC system parameters and coil loads, as well as the energy consumption of primary plant equipment. The list of EnergyPlus features shown in the manual (EnergyPlus, 2010i) is quite long, but some of the most important are:

- Integrated, simultaneous solution,
- Heat balance based solution technique for building thermal loads,
- Transient heat conduction through building elements,
- Improved ground heat transfer modelling,
- Combined heat and mass transfer model,
- Thermal comfort models,
- Advanced fenestration calculations,
- Daylighting controls,
- Loop based configurable HVAC systems, etc.

Detailed literature survey on EnergyPlus validation is presented in Appendix B.

The building energy consumption depends on both building and HVAC system characteristics but also on their interaction. The choice of an HVAC system directly affects the life-cost of the building and indirectly influences the environment. As already stated, the performance of the following HVAC systems have been investigated: variable air volume system (VAV), constant air volume system (CAV), fan-coil system with dedicated air (FC), chilled ceiling system with embedded pipes, dedicated air and radiator heating (EMB), and chilled ceiling system with exposed aluminium panels, dedicated air and radiator heating (ALU). Each of simulated systems was coupled with 3,840 office building models as originally presented in Figure 4.1. The results of annual energy consumptions were normalised per floor area.

In order to confirm that the HVAC systems in each of the simulated building models were properly sized to meet the zone setpoint an analysis of the annual number of hours when the setpoint was not met was conducted. Table 4.1 presents the average number of hours per year when the heating/cooling setpoint was not met during the occupied hours by each of the five studied HVAC systems. It can be seen that the average values are very small. This confirms that the HVAC systems were sized properly and that they are capable of providing the desired zone conditions during occupied hours in each of 3,840 simulated office building models.

Table 4.1. Average annual number of hours heating/cooling setpoint not met during occupied period

System	Setpoint not met while occupied [h/yr]	
	Heating	Cooling
VAV	26.94	7.68
CAV	23.62	10.59
FC	32.23	2.09
EMB	31.96	19.17
ALU	31.94	21.87

The analysis of HVAC systems simulation outputs were conducted in two parts. The first part presents the analysis of office buildings energy requirements when coupled with different HVAC systems. On the other hand, the main aim of the second part of analysis it to generate mathematical models which can predict cooling, heating and auxiliary energy consumptions of particular HVAC system as a function of building demands.

4.1. Office buildings energy end-use requirements

Energy requirements of HVAC systems can be classified into three categories:

- Cooling energy requirements,
- Heating energy requirements, and
- Auxiliary energy requirements.

Simulation results of five HVAC systems coupled with office building models analysed in this research were also subdivided into these three categories.

Cooling, heating and auxiliary energy requirements of different HVAC systems coupled with office building stock were compared among themselves, as well as with the results of the building demands calculation (System "0" – see 3.3.1). Basic statistical parameters, such as mean, standard deviation, maximum, minimum, etc., were used for the comparison.

4.1.1. Cooling energy requirements

The overview of cooling energy requirements for the analysed HVAC systems and associated statistical analysis presented in Table 4.2 is based on the results obtained when each of these HVAC system types were coupled with 3,840 different office building models.

Table 4.2. Summary of cooling energy requirement for office buildings depending on their HVAC system type

	Cooling energy requirement [kWh/m^2/yr]					
	System "0"	**VAV**	**CAV**	**FC**	**EMB**	**ALU**
Mean	21.55	16.42	12.67	30.72	33.67	28.54
Std. Deviation	13.327	6.562	5.325	18.324	18.777	15.655
Minimum	1.90	4.39	2.97	3.52	2.21	2.22
Maximum	71.99	37.42	30.59	100.15	98.00	84.34
Median	18.28	15.15	11.59	26.32	30.29	25.49
Percentile 25	11.77	11.56	8.64	17.15	19.50	17.07
Percentile 75	28.04	20.17	15.63	39.83	44.52	36.86

By comparing means, the results indicate that buildings equipped with all-air systems (VAV and CAV) in general have lower cooling requirements primarily due to extensive use of free cooling. CAV systems perform slightly better in comparison with the VAV system mainly due to higher free cooling availability, since the supply air setpoint is varied, while in the VAV system the supply air setpoint is fixed to 16°C. Buildings with fan-coil systems (FC) demand more cooling energy, mainly due to the FC system operates with a constant minimum outdoor air supply, sufficient only to cover fresh air requirements, which means that the system is not able to utilise free cooling as much as all-air systems. Free cooling availability in all-air systems diminishes the impact of additional heat gains from auxiliary equipment, in particular fans, on the cooling requirements. In the systems with limited free cooling availability, this impact is quite substantial, despite the fact that these systems have much lower additional heat gains from auxiliary equipment when compared to the all-air systems

(Table 4.6). This became obvious when comparing systems cooling requirements with the buildings cooling demands (System "0").

Buildings equipped with chilled ceiling systems, whether with embedded pipes (EMB) or exposed aluminium panels (ALU), perform similar to buildings with fan-coil systems, although the cooling requirements of systems with aluminium panels are slightly lower. The reason why the system with embedded pipes is slightly less efficient when compared to system with aluminium panels is because part of cooling energy is being absorbed by the concrete ceiling. From an energy balance point of view, it might be expected that systems with chilled ceiling have lower cooling energy requirements, since the room setpoint is increased by 2°C, which results in decreased ventilation and transmission losses. However, that is not the case most likely due to the mechanism of how these systems respond to cooling demands. The zone air is maintained at desired setpoint by natural convection between the air and surrounding surfaces. The chilled ceiling, which is at a lower temperature than the zone air, cools the air by convection, but the most of cooling energy is radiated to surrounding surfaces. The temperature of the surrounding surfaces is decreased, which helps in conditioning the zone air. However, during that process, part of the cooling energy is also being absorbed by these surfaces, increasing the cooling energy consumption.

Figure 4.2 shows the frequencies of each HVAC system cooling energy requirements as well as building cooling demands. The chart at the top of Figure 4.2 shows different office buildings cooling demands. The results presented in Figure 4.2 clearly indicate that the pattern of energy consumptions differs significantly depending on the system type. All-air systems (VAV and CAV) have narrower cooling requirements distribution, while FC, EMB and ALU systems have a longer tail and the lower maximum frequency.

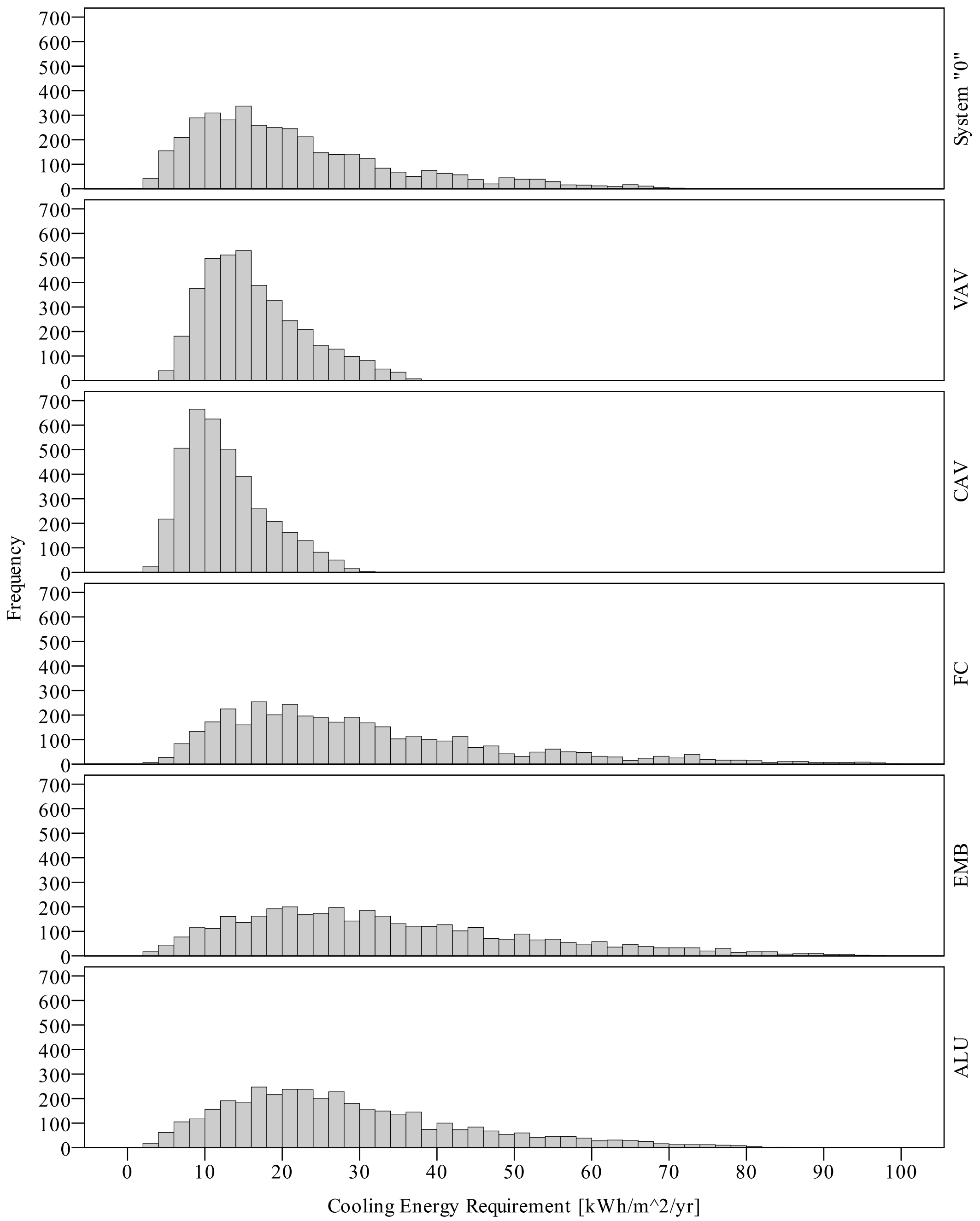


Figure 4.2. Cooling energy requirement histograms

Following the analogy with the primary systems, where the performance of energy generating equipment can be defined by using a single coefficient, such as boiler efficiency or chiller coefficient of performance, the secondary HVAC system seasonal efficiency was calculated and analysed in this work to investigate the possibility of determining the HVAC system energy requirements as a function of a single parameter.

The system seasonal efficiency is defined in a similar way to the SBEM definition of HVAC system seasonal heating and cooling efficiencies (DCLG, 2011). The SBEM defines the "System Seasonal Efficiency for Heating" (SSEFF) as the ratio of the total heating demand in zones served by a system to the energy input into the heat generating equipment. The HVAC system cooling efficiency is defined by using the term "System Seasonal Energy Efficiency Ratio" (SSEER) as the ratio of total cooling demand in zones served by an HVAC system to the energy input into the cold generator. Both parameters take into account the efficiency of primary systems, heat gains and losses of the distribution system, etc. but exclude the energy associated with fans, pumps and controls (auxiliary energy). Also, the building total heating and cooling demands, used in abovementioned coefficients, are calculated for idealised conditions.

The secondary HVAC system seasonal efficiency defined in this research differs from the SBEM coefficients SSEFF and SSEER in that the impact of the primary systems (heat and cold generating equipment) on the system seasonal efficiency is excluded from the calculations. The secondary HVAC system seasonal efficiency is defined as the ratio of the annual building demand (heating or cooling) to the annual energy input to the HVAC secondary system. The annual building demand, either heating or cooling, is the sum of all building zone demands which are calculated assuming that zones are served by an idealised air system (the outputs from the "System 0" simulations). The overall system seasonal efficiency (equivalent to SBEM coefficients SSEFF and SSEER) can be determined by multiplying primary system seasonal efficiency with the HVAC system seasonal efficiency.

The secondary HVAC system seasonal cooling efficiency was calculated as a ratio of the annual building cooling demand and the HVAC system cooling energy requirement for the same building (equation).

$$\boldsymbol{HVAC\ System\ Seasonal\ Cooling\ Efficiency =}$$
$$= \frac{\boldsymbol{Annual\ Building\ Cooling\ Demand}}{\boldsymbol{HVAC\ System\ Cooling\ Energy\ Requirement}} \qquad \textbf{4.1}$$

Table 4.3 presents statistical parameters of HVAC systems seasonal cooling efficiency data set. It can be seen that under no circumstances the FC, EMB and ALU systems require less energy than building cooling demand (maximum seasonal cooling efficiency is less than one). In contrast, the VAV and CAV systems in most cases have seasonal cooling efficiency above one, mainly due to usage of free cooling.

Table 4.3. HVAC system seasonal cooling efficiency

	Seasonal cooling efficiency				
	VAV	**CAV**	**FC**	**EMB**	**ALU**
Mean	1.21	1.58	0.69	0.63	0.74
Std. Deviation	0.291	0.367	0.028	0.064	0.072
Minimum	0.43	0.63	0.52	0.47	0.54
Maximum	1.96	2.63	0.75	0.90	0.92
Median	1.21	1.59	0.70	0.64	0.74
Percentile 25	1.00	1.32	0.68	0.60	0.69
Percentile 75	1.42	1.83	0.71	0.67	0.79

Histograms in Figure 4.3 show distributions of HVAC systems seasonal cooling efficiencies. CAV and VAV systems are almost normally distributed across wide range of efficiencies. Chilled ceiling systems have much lower range of seasonal cooling efficiencies, but still quite wide to be presented as a single parameter. Only FC system shows uniform efficiency across wide range of simulated buildings. 50% of all FC systems have seasonal cooling efficiency between 0.68 and 0.71.

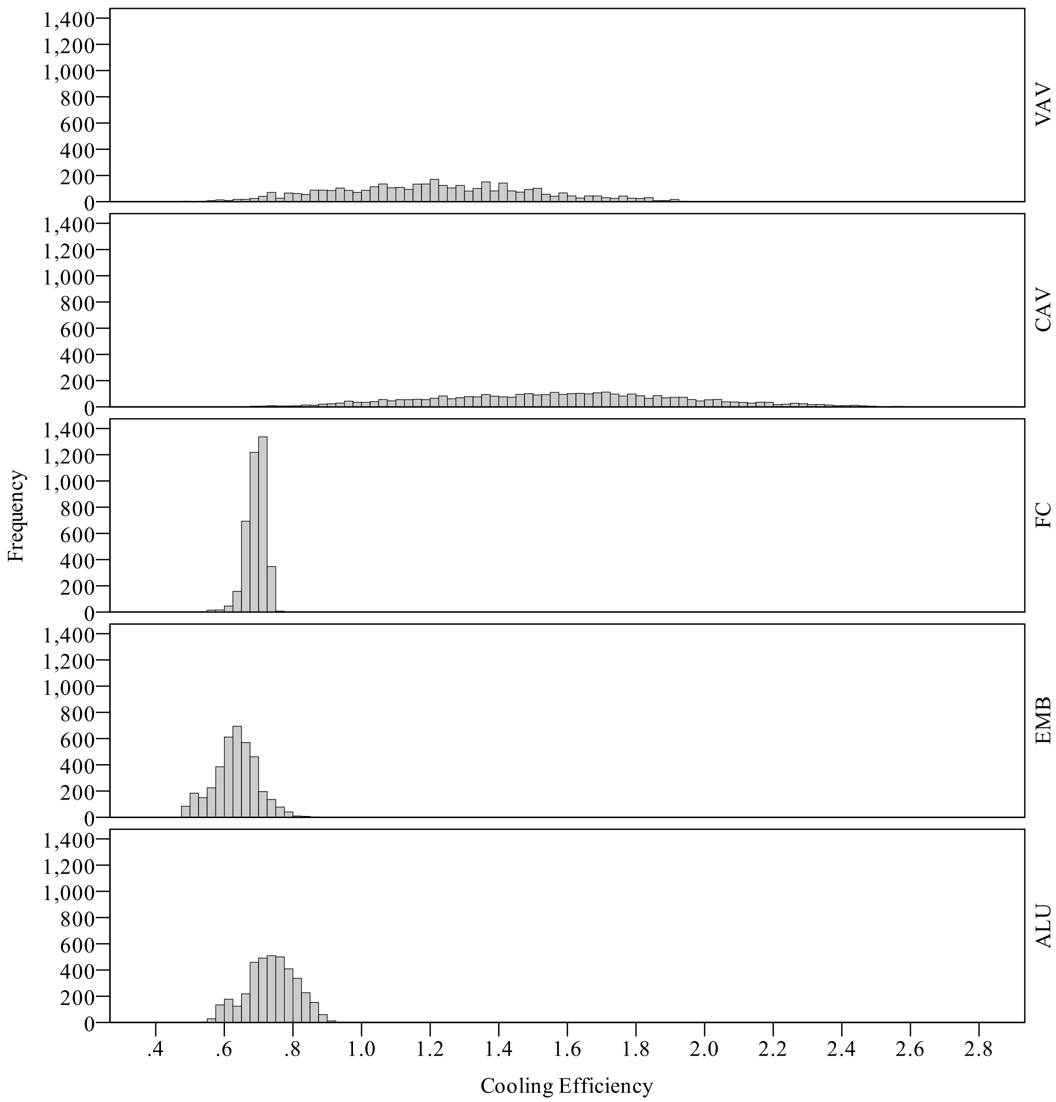


Figure 4.3. HVAC system seasonal cooling efficiency histograms

Scatter plots in Figure 4.4 present the relationship between studied HVAC systems cooling requirements and associated buildings cooling demands. The difference in energy consumption of various HVAC systems when coupled with the same building is noticeable. In addition, it can be seen that there is a pattern in systems cooling requirements when plotted against cooling demands. Beside the FC systems, which cooling requirements seems to have linear relation to the building demands, all other systems show non-linear dependency. The black line in diagram splits the dataset into two zones of seasonal cooling efficiencies as defined by equation 4.1: above and below 1. The systems located above the line have efficiency below one, while the systems below the line have efficiency above one. Although in general VAV and CAV systems have higher seasonal cooling efficiencies, the certain number of buildings equipped with these systems has seasonal cooling efficiency below one (25% and 6% of all simulated buildings in VAV and CAV systems respectively). This occurs in buildings with low cooling demand where the free cooling availability is reduced, mainly in buildings with the building fabric type 1 and 2 (buildings with minimum insulation or no insulation at all).

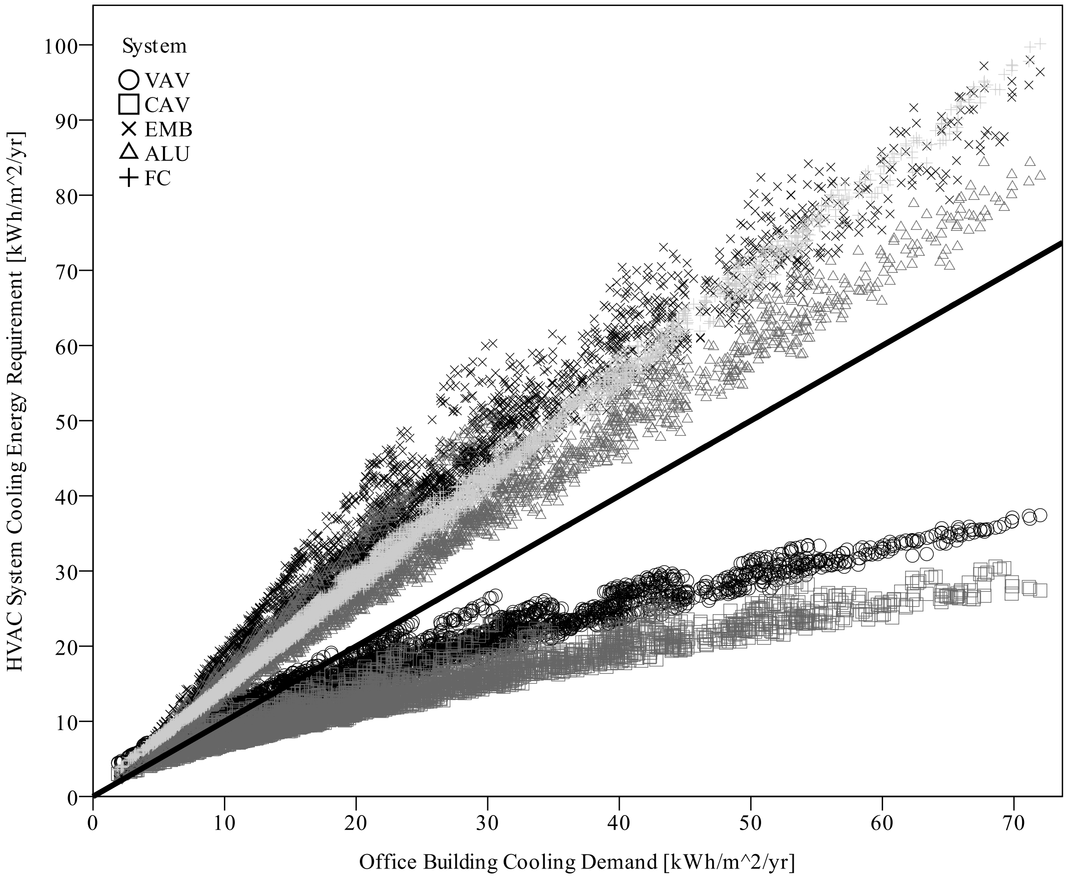


Figure 4.4. HVAC system cooling energy requirement vs. building cooling demand

4.1.2. Heating energy requirements

The overview of heating energy requirement for analysed HVAC systems and associated statistical analysis is presented in Table 4.4.

Table 4.4. Summary of heating energy requirement for office buildings depending on their HVAC system type

	Heating energy requirement [kWh/m²/yr]					
	System "0"	**VAV**	**CAV**	**FC**	**EMB**	**ALU**
Mean	48.91	43.19	35.72	26.63	27.39	26.98
Std. Deviation	26.305	24.415	22.658	21.290	17.403	17.278
Minimum	18.97	13.80	7.86	5.15	8.58	8.55
Maximum	117.79	113.27	96.66	89.36	81.34	80.68
Median	40.03	34.79	27.57	18.34	20.94	20.47
Percentile 25	29.46	24.99	19.73	11.31	14.38	14.12
Percentile 75	54.53	49.54	41.17	30.41	32.49	31.89

In contrast to the cooling requirements, the average heating energy requirement of each group of buildings equipped with the one particular HVAC system is smaller than average building heating demand (System 0, Table 4.4). The highest difference between these means is for HVAC systems which operate with a minimum fresh air intake (fan-coil system and both chilled ceiling systems), primarily due to installed heat recovery unit in the supply air stream. The relatively high effectiveness of a heat recovery unit (65%) affects the ventilation losses which are significantly reduced. The other two systems, CAV and VAV, have higher heating energy requirements than systems with a dedicated air, although CAV system performs slightly better when compared to the VAV system, mostly due to two reasons. Namely, being controlled by a variable supply air temperature setpoint, the CAV system has decreased zone reheating requirements, which is particularly important in situations when some of thermal conditioned zones have cooling requirements while other zones request heating. Second reason is additional heat gains from auxiliary equipment, predominantly supply air fan, which is larger in CAV systems since they work with a maximum constant air flow rate all the time. Opposite to this, VAV systems decrease air flow rate at reduced demand and preserve energy in this way.

Histograms presented in Figure 4.5 show the distribution of each HVAC systems heating energy consumption as well as office buildings heating demands. As already mentioned, buildings with minimum insulation (BF1) have significantly higher heating requirements which separates them from all other buildings. This is obvious as histograms for each system are split into two groups of results, the right one groups buildings with BF1 level of insulation and the left all others.

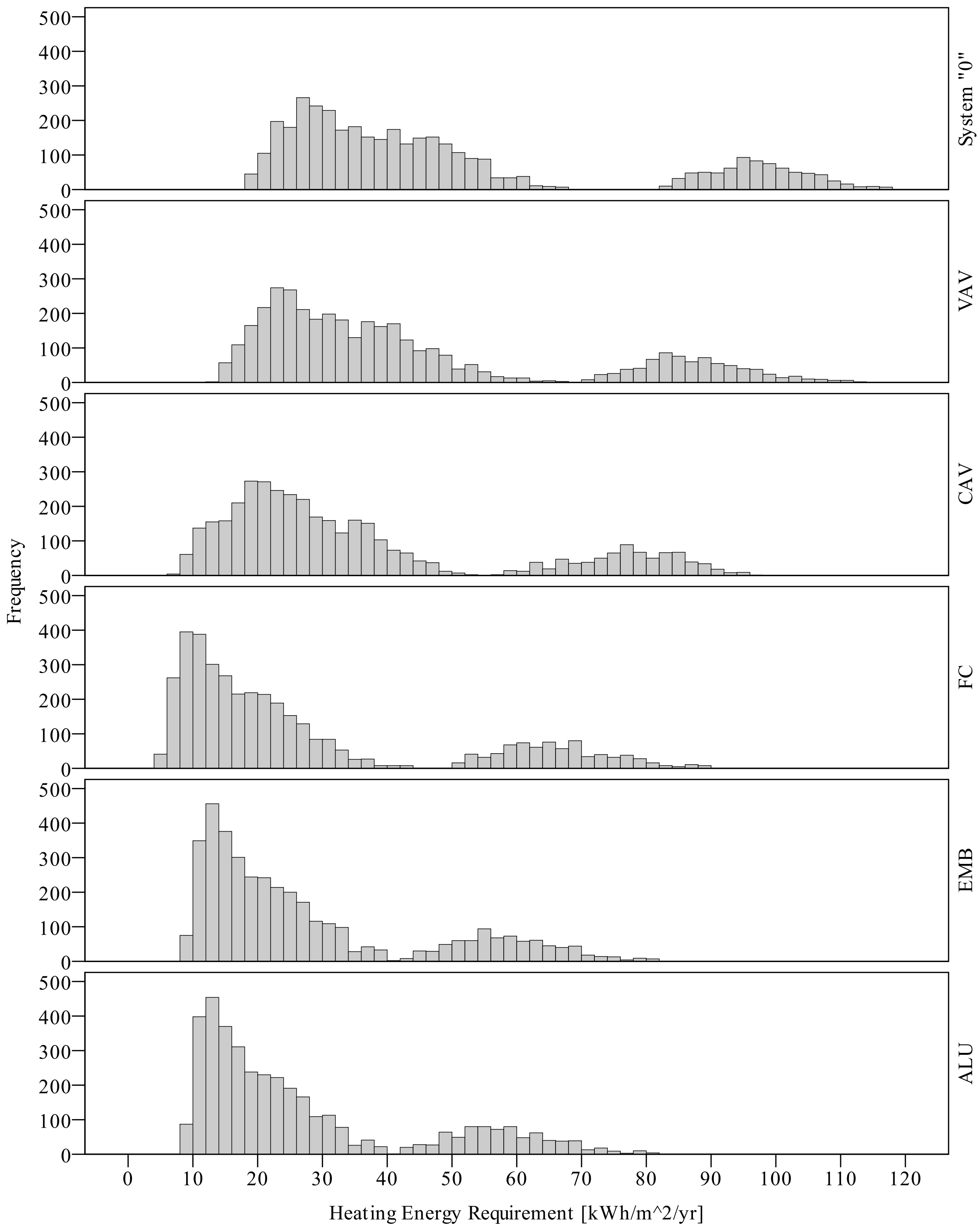


Figure 4.5. Heating energy requirement histograms

Statistical analysis on secondary HVAC systems seasonal heating efficiencies calculated by equation 4.2 and presented in Table 4.5 confirms previous statement that almost all systems have lower heating requirements than participating building heating demands. There are some exceptions in the VAV system where the minimum seasonal heating efficiency is below one.

$$HVAC\ System\ Seasonal\ Heating\ Efficiency = \\ = \frac{Annual\ Building\ Heating\ Demand}{HVAC\ System\ Heating\ Energy\ Requirement} \qquad \mathbf{4.2}$$

Table 4.5. HVAC system seasonal heating efficiency

	Seasonal heating efficiency				
	VAV	**CAV**	**FC**	**EMB**	**ALU**
Mean	1.15	1.47	2.20	1.89	1.92
Std. Deviation	0.106	0.264	0.549	0.215	0.229
Minimum	0.65	1.14	1.31	1.45	1.46
Maximum	1.40	2.70	3.72	2.32	2.41
Median	1.16	1.39	2.17	1.88	1.90
Percentile 25	1.12	1.27	1.74	1.71	1.74
Percentile 75	1.22	1.58	2.61	2.06	2.09

The histogram for VAV systems in Figure 4.6 shows that the number of cases where the minimum seasonal heating efficiency is below one is rather small (330 cases, which is less than 10%). Another clear conclusion from the histograms is that the range of seasonal heating efficiencies for the VAV system is the narrowest. 50% of all cases have seasonal heating efficiency between 1.12 and 1.22. Although the VAV system has the smallest range of seasonal heating efficiencies when compared to other systems, it is still much larger than the range of seasonal cooling efficiencies of FC system which grouped 50% of all data in 0.03 increment range. All other systems have significantly wider range of seasonal heating efficiencies for different building scenarios, which leads to conclusion that the unique HVAC system seasonal heating efficiency cannot be determined for the whole office building stock.

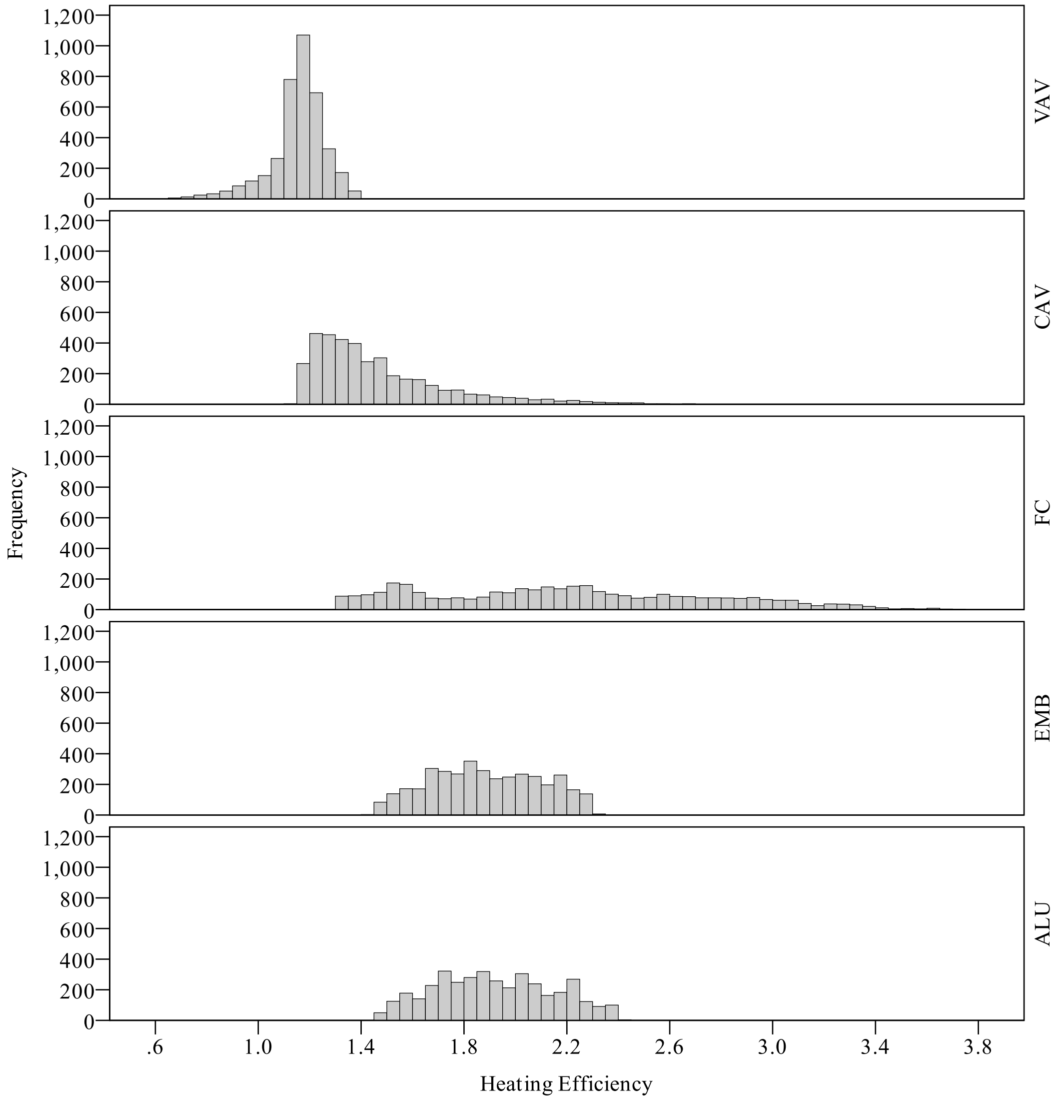


Figure 4.6. HVAC system seasonal heating efficiency histograms

Figure 4.7 visualises the relationship between HVAC system heating requirements and office buildings heating demands. The most notable from the graph is the split of each of FC system and chilled ceiling systems datasets into three groups. The separation is related to the building floor space arrangement. Bottom line in all three system types groups the buildings with the open plan space arrangement (building types one and three). The group of buildings with a highest heating energy consumption is associated with building type two (cellular daylit building type) which floor space is divided by a corridor into two cellular office areas. Due to different area orientation, it happens that one zone requires heating, while at the same time the opposite zone has a cooling demand. Open plan office area at the same time might have neither heating nor cooling requirements. The group of buildings with heating requirements between these two datasets belong to the building type four, which combines cellular offices and open plan space. Both VAV and CAV systems are also affected by floor space arrangement, but not as obvious as other systems. Another noticeable thing which requires explanation is that the uniformly lower heating requirements of the FC system, in comparison to chilled ceiling systems, becomes higher for buildings with minimum level of insulation, despite FC system has higher additional heat gains from auxiliary equipment. The answer is in the additional very thin layer of insulation placed above pipes in chilled ceiling systems. This particularly affects the heat loss through roof element. While in insulated buildings this additional layer does not have significant impact on heating requirements, in the buildings with no insulation at all, which is the case in buildings with the BF1 level of insulation, the impact is significant.

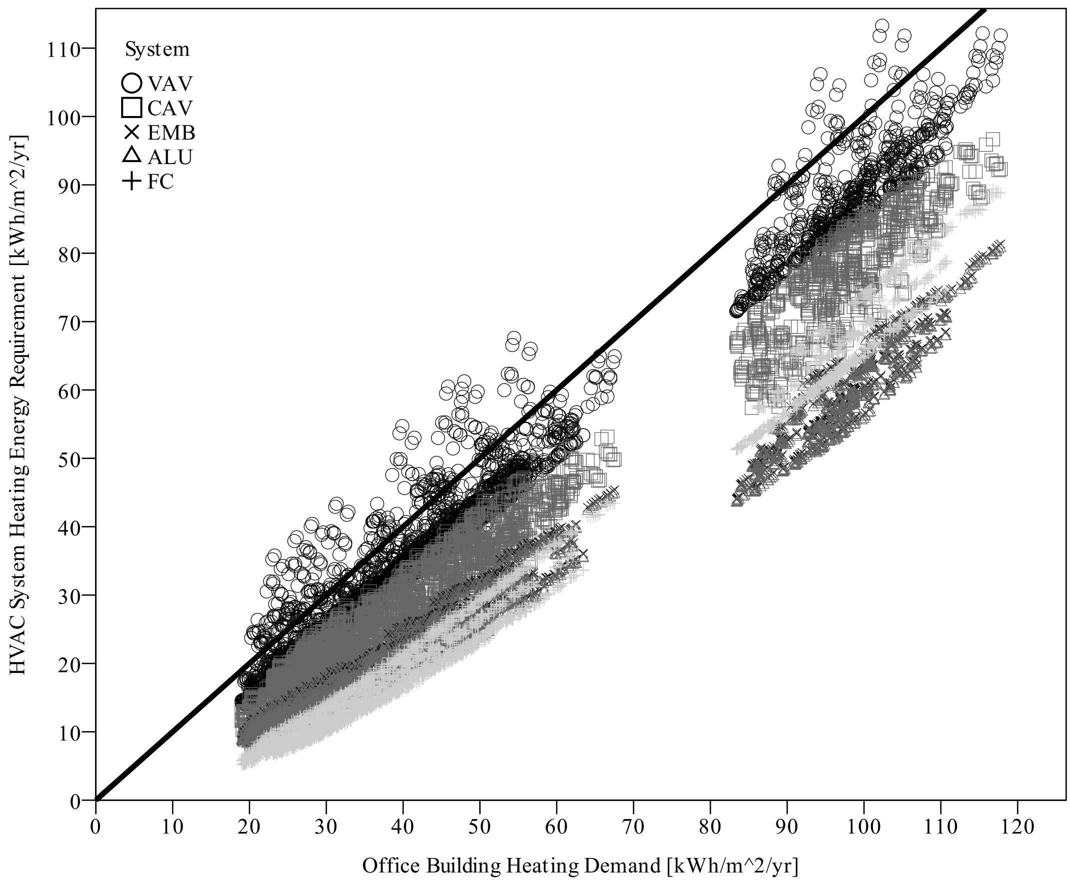


Figure 4.7. HVAC system heating energy requirement vs. building heating demand

4.1.3. Auxiliary energy requirements

HVAC systems auxiliary energy consumptions are often overlooked when the building energy end use is analysed, although sometimes fans and pumps can be the dominant energy consumer and they should not be excluded from analyses. The summary of auxiliary energy consumptions of HVAC systems studied in this research is presented in Table 4.6.

Table 4.6. HVAC system auxiliary energy requirement in office buildings for different HVAC system types

	Auxiliary energy requirement [kWh/m²/yr]				
	VAV	**CAV**	**FC**	**EMB**	**ALU**
Mean	13.50	26.46	11.34	7.73	7.63
Std. Deviation	4.605	9.099	2.966	1.431	1.380
Minimum	5.06	9.89	5.58	4.58	4.59
Maximum	28.40	56.97	20.25	12.53	12.44
Median	12.66	24.77	10.91	7.61	7.53
Percentile 25	9.87	19.25	9.06	6.79	6.75
Percentile 75	16.24	31.82	13.14	8.57	8.44

The system with the highest auxiliary energy consumption is, as expected, the CAV system. Its average auxiliary energy consumption is twice as high as that of the VAV system. Fan-coil system requires slightly less energy for auxiliary equipments when compared to the VAV system, while the chilled ceiling systems have the smallest consumption. The auxiliary energy consumption depends on the building demands, beside the systems type, and the range of consumptions for each system is quite wide. The ratio between maximum and minimum auxiliary energy consumption goes from three in case of chilled ceiling systems up to almost six for all air systems.

All-air systems, as well as the fan-coil system, use forced convection to transfer energy from the hot/chilled water to the air which means that it is expected that the most of auxiliary energy is used by fans. On the other hand, chilled ceiling systems use fans only to deliver fresh air while the air conditioning is based primarily on natural convection and radiation, which should result in higher pump energy consumption.

Figure 4.8 presents shares of fans and pumps energy consumptions in total auxiliary energy consumption. It can be seen that fans are dominant auxiliary energy consumers in CAV, VAV and FC systems.

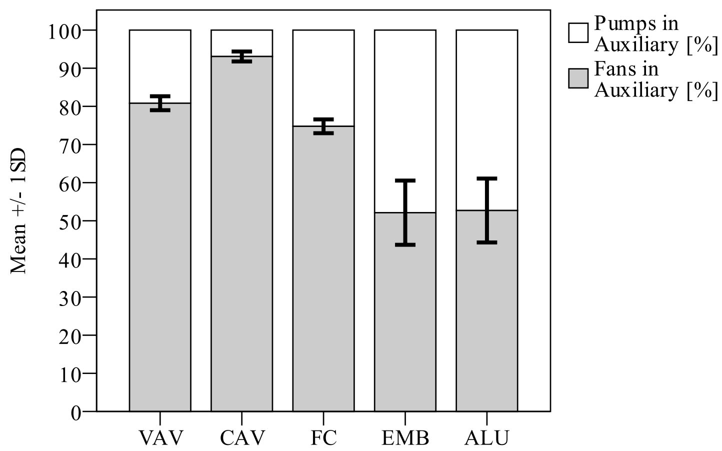


Figure 4.8. Fans/pumps share in auxiliary energy requirements

Table 4.7 provides additional information about fans share in auxiliary energy requirements. On average, fans consume 93%, 80% and 75% of total auxiliary energy consumption in CAV, VAV and FC systems respectively, while the variation between the minimum and the maximum share is quite low, less than 10%. In the 50% of all cases, the variation is below 3%. The share ratio is quite different in chilled ceiling systems. The average fans share is slightly above 50%, however, the deviation is rather high. In some cases, usually when the total building demands are small, fans might consume up to 67% of total auxiliary energy consumption, while in building with higher demand the pumps share in auxiliary consumption can be as much as 75%.

Table 4.7. Fan energy requirements share in total auxiliary energy requirements for different HVAC systems in office buildings

	Fans share in auxiliary energy requirements [%]				
	VAV	**CAV**	**FC**	**EMB**	**ALU**
Mean	80.82	93.08	74.77	52.12	52.69
Std. Deviation	1.826	1.303	1.811	8.420	8.376
Minimum	75.66	89.13	68.70	28.76	29.00
Maximum	85.35	96.56	79.10	70.90	70.75
Median	80.83	93.10	74.78	52.94	53.69
Percentile 25	79.53	92.20	73.53	46.45	47.16
Percentile 75	82.15	94.00	76.11	58.35	58.91

Similar to the HVAC systems seasonal cooling and heating efficiencies, the auxiliary energy consumption is divided by building total demand in order to investigate the correlation between them as presented in equation 4.3. Total building demands are composed of heating and cooling demands.

$$\boldsymbol{HVAC\ System\ Auxiliary\ Share} = \\ = \frac{\boldsymbol{HVAC\ System\ Auxiliary\ Energy\ Requirement}}{\boldsymbol{Annual\ Building\ Total\ (heating\ and\ cooling)\ Demand}} \qquad \textbf{4.3}$$

The summary results presented in Table 4.8 show that the auxiliary energy consumption share in total building demands of each individual HVAC system varies widely across office building stock. This leads to the conclusion that auxiliary energy consumption cannot be calculated by using a single coefficient, which would multiply total building demand. The error in prediction would be particularly large in all-air systems where the energy required by auxiliary equipment ranges from 10% up to 60% of total building demands.

Table 4.8. HVAC system auxiliary energy consumption share in total building demands

	Auxiliary share in building total demands				
	VAV	**CAV**	**FC**	**EMB**	**ALU**
Mean	0.20	0.39	0.17	0.12	0.12
Std. Deviation	0.052	0.101	0.032	0.031	0.031
Minimum	0.08	0.16	0.09	0.05	0.05
Maximum	0.32	0.63	0.22	0.18	0.18
Median	0.20	0.40	0.17	0.12	0.12
Percentile 25	0.16	0.32	0.14	0.10	0.10
Percentile 75	0.24	0.46	0.19	0.14	0.14

While both HVAC system heating and cooling energy requirements are in strong correlation to buildings heating and cooling demand respectively, as it can be seen from Figure 4.4 and Figure 4.7, the HVAC systems auxiliary energy consumptions do not correlate well with building total demands (Figure 4.9).

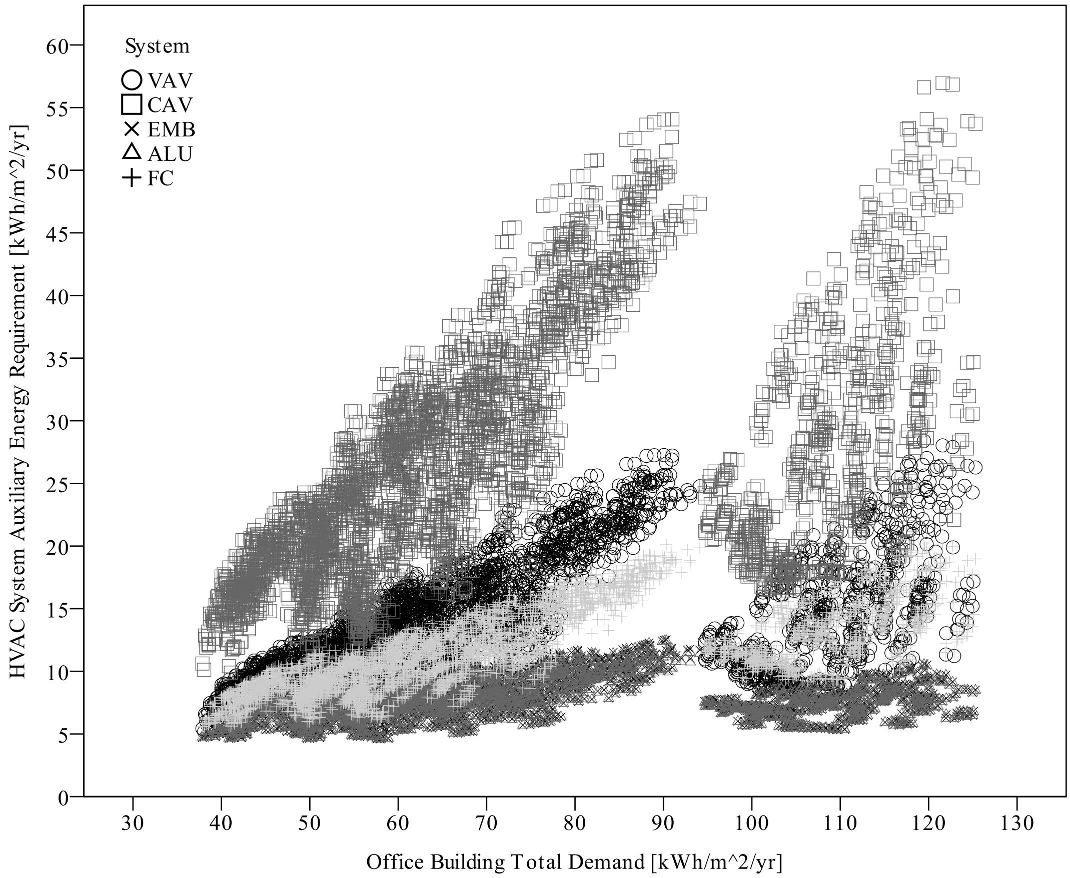


Figure 4.9. HVAC system auxiliary energy requirement vs. buildings total demand

4.2. HVAC systems energy requirements in office buildings

Previous chapters described characteristics of the office building stock when coupled with different HVAC systems. Basic statistical parameters were used to describe cooling, heating and auxiliary energy requirements of the whole set of office building models coupled with particular HVAC system. In the following sections, each of these couplings is further analysed with the main aim to explore possibility of representing HVAC system energy consumption as a function of a building demand. Building demand has been chosen as an independent variable since that is a building characteristic which all systems have in common. Relationship between HVAC system energy consumption and building demand has been explored by using regression analysis.

Besides regression analysis (either single or multiple) there are a number of different data-driven modelling methods which are suitable for studies in which both input and output variables are known. A summary of such data-driven modelling methods is given in the ASHRAE Fundamentals Handbook Chapter 19 (2009) and this discusses advantages and disadvantages as well as the basic characteristics of various data-driven modelling methods including change-point models, the variable-base degree-day method, data-driven bin method, Fourier series analysis, artificial neural networks, to mention just a few. However, the preference in this research was given to single and multiple regression modelling methods. There are two reasons for this. Firstly, the limited number of independent variables used for the model development (only two: heating and cooling demands). Secondly, they are simpler when compared to other data driven modelling methods. For example, the data-driven bin model and Fourier series analysis were classified as moderately difficult, while the artificial neural networks were considered as a complex modelling method according to the ASHRAE Fundamentals Handbook Chapter 19 (2009). This is very important since model simplicity is one of two the most important requirements specified in the scope of this research. The second requirement is model accuracy.

4.2.1. Regression models

The main aim of regression analysis is to determine as simple as possible model which fits the observed data well enough (EnergyPlus simulation outputs) and calculates the HVAC system energy requirements correctly. The goal is not to find the model that calculates the requirements perfectly, if that is possible at all, since such a model would probably have too many parameters to be useful.

A regression model is an equation which defines dependent variable as a function of one or more independent variables, and one or more model parameters. In this particular regression analysis, only two independent variables were used: building cooling demand and building heating demand, which are actually the EnergyPlus outputs from the System”0” simulations. HVAC systems heating, cooling and auxiliary energy requirements, also calculated by EnergyPlus, were used to fit regression models.

Several mathematical functions were chosen for a regression analysis and they were subdivided into two sets. The first set is composed of models based on a single independent variable, while models with two independent variables are grouped into second set. A polynomial function was used as a basis for both sets. Regression models with one independent variable were based on the second order polynomial function presented in equation 4.4.

$$\boldsymbol{y(x) = a + b \cdot x + c \cdot x^2} \qquad \textbf{4.4}$$

The simplest model based on equation 4.4 is a linear model which pass through the origin (x = 0, y = 0). In the following text this model will be referred as LIN1. It has been selected for analysis since this model represents the HVAC system seasonal efficiency. The system seasonal efficiency was described previously and it has been shown that it is not a proper parameter to be used for determining HVAC system requirements as a function of building demands. The more complex models will also refer to this model in order to show its inaccuracy. Second model is also linear, but with an interceptor which differs from zero (LIN2). The last model is a full quadratic model (QUAD). A summary of polynomial single argument models are presented in Table 4.9.

Table 4.9. Single independent variable polynomial models

Model	Model parameters
LIN1	(b) ≠ 0; (a, c) = 0
LIN2	(a, b) ≠ 0; (c) = 0
QUAD	(a, b, c) ≠ 0

In addition to these three models, the power function (POW) was also selected for analysis (equation 4.5). The main reason why the power function was added to analysis was due to quadratic function has the turning point. The turning point in QUAD models might lead to model predictions which are not scientifically sensible, such as negative energy consumption. However, large extrapolations, beyond the range of independent variable, in regression models are not recommendable at all.

$$\boldsymbol{y(x) = a + b \cdot x^c} \qquad \textbf{4.5}$$

The above single independent variable models were used to predict either the HVAC systems cooling energy requirement as a function of the building cooling demand or the HVAC systems heating energy requirement as a function of the building heating demand. Single independent variable models were not used for predicting the HVAC systems auxiliary energy consumption since they are not suitable.

The second group of regression models is based on the second order polynomial function with two independent variables (equation 4.6).

$$\boldsymbol{y(x_1, x_2) = a + b \cdot x_1 + c \cdot x_2 + d \cdot x_1^2 + e \cdot x_1 \cdot x_2 + f \cdot x_2^2} \qquad \textbf{4.6}$$

Independent variables x_1 and x_2 represent building cooling and heating demands respectively. Four models based on equation 4.6 were chosen for analysis and characteristic of these models were presented in Table 4.10

Table 4.10. Two independent variables polynomial models

Model	Model parameters
POLY11	(a, b, c) ≠ 0; (d, e, f) = 0
POLY21	(a, b, c, d, e) ≠ 0; (f) = 0
POLY12	(a, b, c, e, f) ≠ 0; (d) = 0
POLY22	(a, b, c, d, e, f) ≠ 0

The main task of the regression analysis is to calculate models best fit parameter values (parameters a, b, c, d, e and f in the above equations). Having determined these values, it is important to evaluate them as well as how close the models fit come to the observed data.

The model best fit parameter values are evaluated by inspecting the 95% confidence interval (or the standard error from which the 95% confidence interval is calculated). If the confidence interval is reasonably narrow, it can be concluded that the best fit parameter values are determined with a reasonable certainty.

How close the predicted values are to the observed data is usually evaluated by checking either the residual sum of squares (RSS) or the coefficient of determination (R^2). Residuals are defined as the difference between the value predicted by the model (y_i) and the associated observed value ($\hat{y}_i$), as presented in equation 4.7.

$$\boldsymbol{e_i = y_i - \hat{y}_i} \qquad \textbf{4.7}$$

Residual sum of squares is the sum of the squares of these differences (equation 4.8). A smaller value of the residual sum of squares means that the predictions are closer to the observed data.

$$\boldsymbol{RSS} = \sum_{i=1}^{N} \boldsymbol{e_i^2} \qquad \textbf{4.8}$$

Another parameter which quantifies the goodness of fit is the coefficient of determination R^2. The R^2 is computed from the residual sum of squares and the total sum of squares (TSS), as it can be seen from equation 4.9. The total sum of squares represents the sum of squared differences between the observe data points and the mean of the whole dataset (equation 4.10).

$$\boldsymbol{R^2} = \boldsymbol{1} - \frac{\boldsymbol{RSS}}{\boldsymbol{TSS}} \qquad \textbf{4.9}$$

$$\boldsymbol{TSS} = \sum_{i=1}^{N} (\hat{\boldsymbol{y}}_i - \bar{\boldsymbol{y}})^2 \qquad \textbf{4.10}$$

A higher value of the coefficient of determination indicates that the model predictions come close to the observed data. Although very useful, the R^2 should not be used as only criterion for whether a fit is reasonable. It should always be checked together with the model parameters which may have values that make no sense, or their confidence intervals may be very high.

Beside the coefficient of determination and the residual sum of squares, several parameters, which may give a better view of goodness of fit, were also included in the analysis. A root mean square deviation (RMSD) is the square root of the variance of the residuals and it indicates the absolute fit of the model to the observed data (equation 4.11). Maximum residual (e_{max}) and minimum residual (e_{min}) might be useful to see the range of the differences. In addition, the mean absolute difference ($\overline{|e|}$) is computed to show the overall magnitude of the differences between predictions and simulations (equation 4.12).

$$\boldsymbol{RMSD} = \sqrt{\frac{\mathbf{1}}{\boldsymbol{N}}\sum_{\boldsymbol{i}=\mathbf{1}}^{\boldsymbol{N}} \boldsymbol{e}_{\boldsymbol{i}}^{2}} \qquad \mathbf{4.11}$$

$$\overline{|\boldsymbol{e}|} = \frac{\mathbf{1}}{\boldsymbol{N}}\sum_{\boldsymbol{i}=\mathbf{1}}^{\boldsymbol{N}} |\boldsymbol{e}_{\boldsymbol{i}}| \qquad \mathbf{4.12}$$

The scatter plot of the residuals against the independent variable also provides additional information about goodness of fit. The residuals should be randomly scattered above and below the origin line (y = 0). There should not be a large clustering of adjacent residuals that are all above or below the origin.

Since the several mathematical models were chosen for regression analysis, and some of them are more complex than others, they have to be compared to each other in order to select the appropriate one. The starting point in model comparison is to confirm that all models parameter values have sensible best fit. If that is true then the next step is to compare their goodness of fit. If simpler model fit better than more complex model (has lower residual sum of squares) then the choice is clear; the more complex model should be abandoned. However, this is rarely the case. The answer to the question which model should be accepted (not which model is better since in the most cases the more complex model will fit the observed data better) cannot be obtained without further statistical analysis. The F-test, or sometimes called the extra sum of squares test, can be used for related models, where the term related means that one model is a simpler case of the other. This test is based on analysis of the differences between the two models residual sum of squares. If models are not related, then instead of the F-test, either Akaike‟s Information Criterion (AIC) or Schwarz-Bayesian Information Criterion (BIC) should be used. These criterions can also be used to evaluate related models. The difference between AIC and BIC is that the latter is more sensitive to the number of parameters in the model (the penalty for each additional model parameter is larger than in the AIC). However, since there is a large number of degree of freedom (due to large number of observed data and relatively small number of model parameters) in the regression models in this research, in almost all cases the F-test, AIC and BIC recommend the more complex model to be adopted.

In the following section the detailed example of evaluating the outputs from the regression analysis is given for the VAV system only. Since this method is the same for all other systems, only the summary of analysis and major findings are presented in this section, while the complete analysis is provided in the Appendix C.

4.2.2. VAV system

The relationship between studied HVAC systems cooling energy requirements and associated buildings cooling demands was previously presented in Figure 4.4. Figure 4.10 presents only the VAV system cooling energy requirements dependence on the associated building cooling demands.

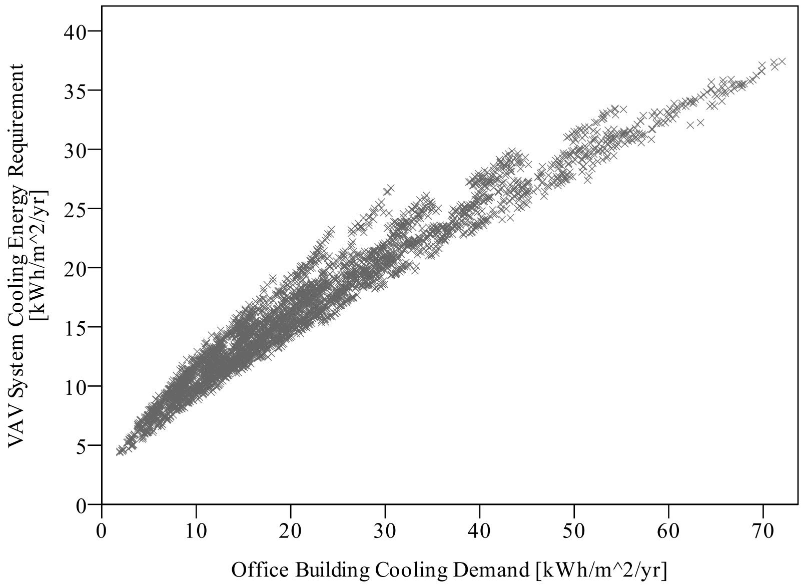


Figure 4.10. VAV system cooling energy requirement as a function of office building cooling demand

These data were firstly fitted to the first set of regression models presented in equations 4.4 and 4.5 (models which are functions of a single independent variable; building cooling demand in this particular case) which resulted in obtaining models best fit parameter values presented in Table 4.11. Standard errors of the estimated best fit parameter values are small and 95% confidence intervals are narrow, providing justification for accepting these parameter values. Model parameter values standard errors and 95% confidence bounds are presented in Table C.1 in the Appendix C.

Table 4.11. Regression model parameter values for single independent variable models of VAV system cooling energy consumption in office buildings

Model	Model Parameters a	b	c
LIN1	-	6.843×10^{-1}	-
LIN2	6.065	4.807×10^{-1}	-
QUAD	4.268	6.498×10^{-1}	-2.879×10^{-3}
POW	1.449	1.829	6.972×10^{-1}

Statistical parameters which describe the goodness of fit of VAV system single variable cooling energy consumption regression models are presented in Table 4.12. Linear model which passes through the origin (LIN1) has the lowest coefficient of determination (≈0.72) and highest residual sum of squares, when compared to other models, which means that the predicted values calculated by this model are furthest from the observed data. Significant improvement in predicting the VAV system cooling energy requirement as a function of a building cooling demand can be made by using the linear model which does not pass through the origin (LIN2). Coefficient of determination of this model is above 0.95, while the root mean square deviation decreased from 3.49 in the case of the LIN1 model to slightly above 1.42. Further improvement can be obtained by both quadratic (QUAD) and power (POW) models. These two models provided the closest fit to observed values, although slightly better prediction is achievable by the power model. Both models have the coefficient of determination above 0.963.

Table 4.12. Comparison of single independent variable models of VAV system cooling energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW
$\bar{y}$	16.42	14.75	16.42	16.42	16.40
σ_y	6.562	9.119	6.406	6.441	6.442
RSS		46830	7757	6041	5992
R^2		0.7167	0.9531	0.9635	0.9638
RMSD		3.4922	1.4213	1.2543	1.2492
e_{max}		6.74	5.96	5.28	5.42
e_{min}		-11.84	-4.27	-2.91	-2.71
$\overline{\lvert e \rvert}$		2.98	1.12	1.00	0.99
$\lvert e \rvert_{95\%}$		5.8	2.73	2.37	2.36

Additional information about regression models goodness of fit can be obtained by inspecting the scatter plot of the residuals against the independent variable, in this case against building cooling demand, and the residuals distribution histograms presented in Figure 4.11. Characteristics of the better model are to have narrower histogram and randomly distributed residuals around the mean. From Figure 4.11 can be concluded that the LIN1 model is not appropriate since there is a large clustering of adjacent residuals both above the mean at the lower end of building cooling demands, and below the mean at the higher end of building cooling demands. Although the LIN2 model provides significant improvement in the VAV system cooling energy requirements predictions, when compared to the LIN1 model, the residuals plot shows that still there is a certain degree of clustering of adjacent residuals. Minor clustering occurs in the area of low building cooling demands, whilst the larger clustering, where all residuals are below the mean, happens in cases with high building cooling demands (roughly above 55 kWh/m2/yr). This was overcome in both quadratic and power model. Residuals are randomly distributed above and below the mean. Residual plots, as well as the residuals distribution histogram shapes, are fairly similar for these two models, which confirm that their predictions are close to each other. In addition, the residuals ranges are thinner than the range in the LIN2 model which makes both power and quadratic model more appropriate for predicting the VAV system cooling energy requirement as a function of building cooling demand.

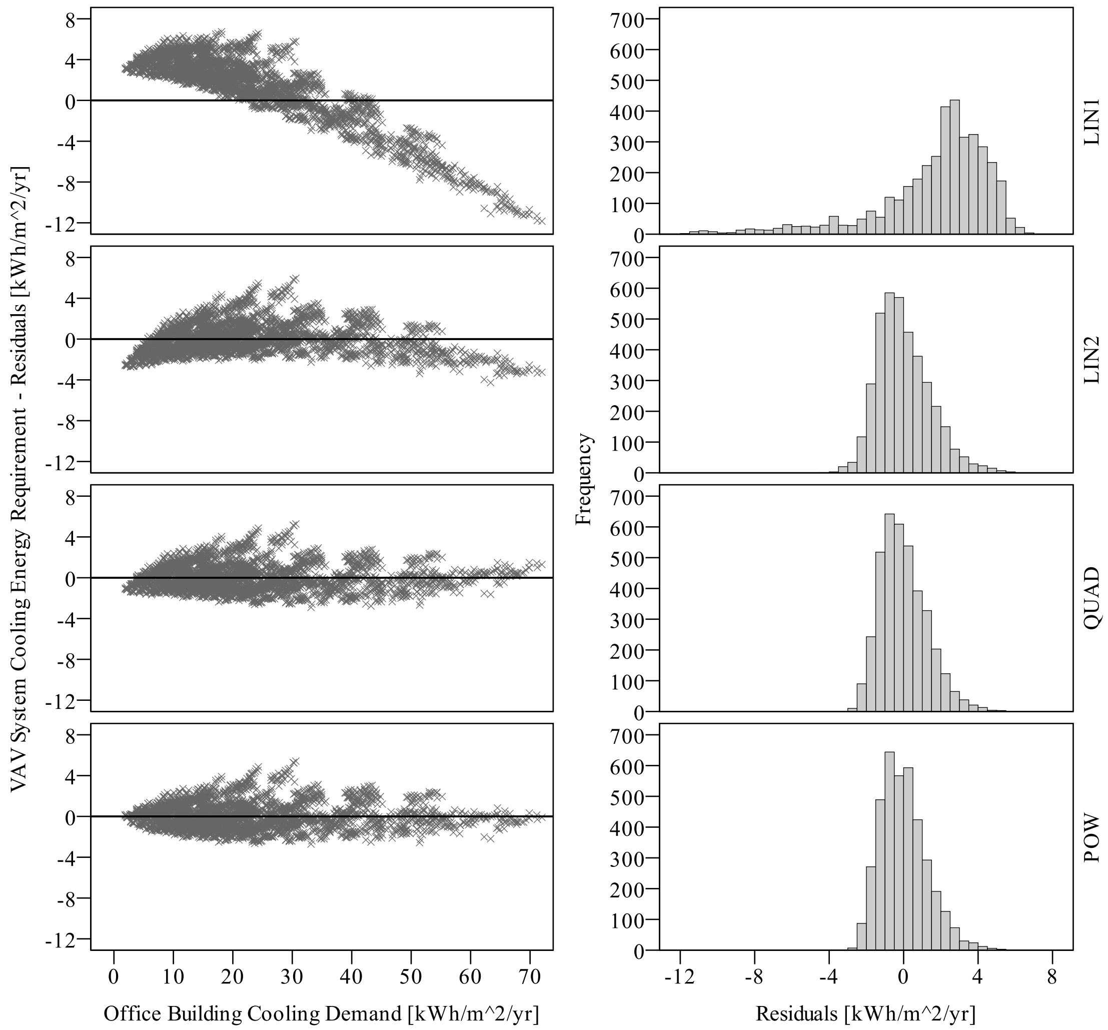


Figure 4.11. Residuals scatter plots and histograms for single independent variable models of VAV system cooling energy consumption in office buildings

The VAV system cooling energy requirements calculated by EnergyPlus were also fitted to the set of polynomial functions based on two independent variables, as previously presented in equation 4.6. Two independent variables are outputs from a building demands calculation; a building cooling demand and a building heating demand. Models best fit parameter values are presented in Table 4.13. Standard errors of the estimated best fit parameter values and 95% confidence bounds are presented in Table C.1 in the Appendix C. These values suggest that the model parameter values can be accepted.

Table 4.13. Regression model parameter values for two independent variables models of VAV system cooling energy consumption in office buildings

Model	Model Parameters					
	a	b	c	d	e	f
POLY11	3.764	5.139×10^{-1}	3.239×10^{-2}	-	-	-
POLY21	3.696	5.128×10^{-1}	-5.272×10^{-3}	-1.839×10^{-3}	3.574×10^{-3}	-
POLY12	4.875	3.732×10^{-1}	5.441×10^{-3}	-	4.461×10^{-3}	-2.096×10^{-4}
POLY22	1.069	5.792×10^{-1}	7.421×10^{-2}	-2.299×10^{-3}	2.628×10^{-3}	-5.091×10^{-4}

Statistical parameters presented in Table 4.14 evaluate the goodness of fit of four polynomial two independent variables regression models. Among these four models, the worst performing one is the first order model (POLY11). However, this model can still provide more accurate prediction of the VAV system cooling energy requirement than the power model, which is the best of single independent variable models. Tremendous improvement can be made by using the second order polynomial functions. Coefficient of determination of each of second order polynomial models is above 0.99 which can be assumed to be almost perfect fit. The best is the POLY22 model, which is based on the full second order polynomial function.

Table 4.14. Comparison of two independent variables models of VAV system cooling energy consumption in office buildings

	Observed	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	16.42	16.42	16.42	16.42	16.42
σ_y	6.562	6.447	6.545	6.538	6.548
RSS		5726	866	1199	695
$\mathbf{R^2}$		0.9654	0.9948	0.9927	0.9958
RMSD		1.2211	0.4749	0.5588	0.4254
e_{max}		4.36	1.42	1.52	1.60
e_{min}		-4.81	-2.28	-2.70	-2.05
$\overline{\lvert e \rvert}$		0.93	0.37	0.44	0.32
$\lvert e \rvert_{95\%}$		2.48	0.90	1.09	0.85

The residuals scatter plots (Figure 4.12), which were plotted against both independent variables, show that the POLY11 model is not appropriate since there is a considerable clustering of adjacent residuals at both ends of building cooling demand scale. All other models show much better distribution of residuals above and below the mean. Since residual plots are very similar for all second order polynomial models, the advantage of the POLY22 model in predicting the VAV system cooling energy requirements become more obvious when compared residuals using distribution histograms (charts at the bottom of Figure 4.12). The POLY22 residuals distribution histogram is the narrower and has the highest maximum bin.

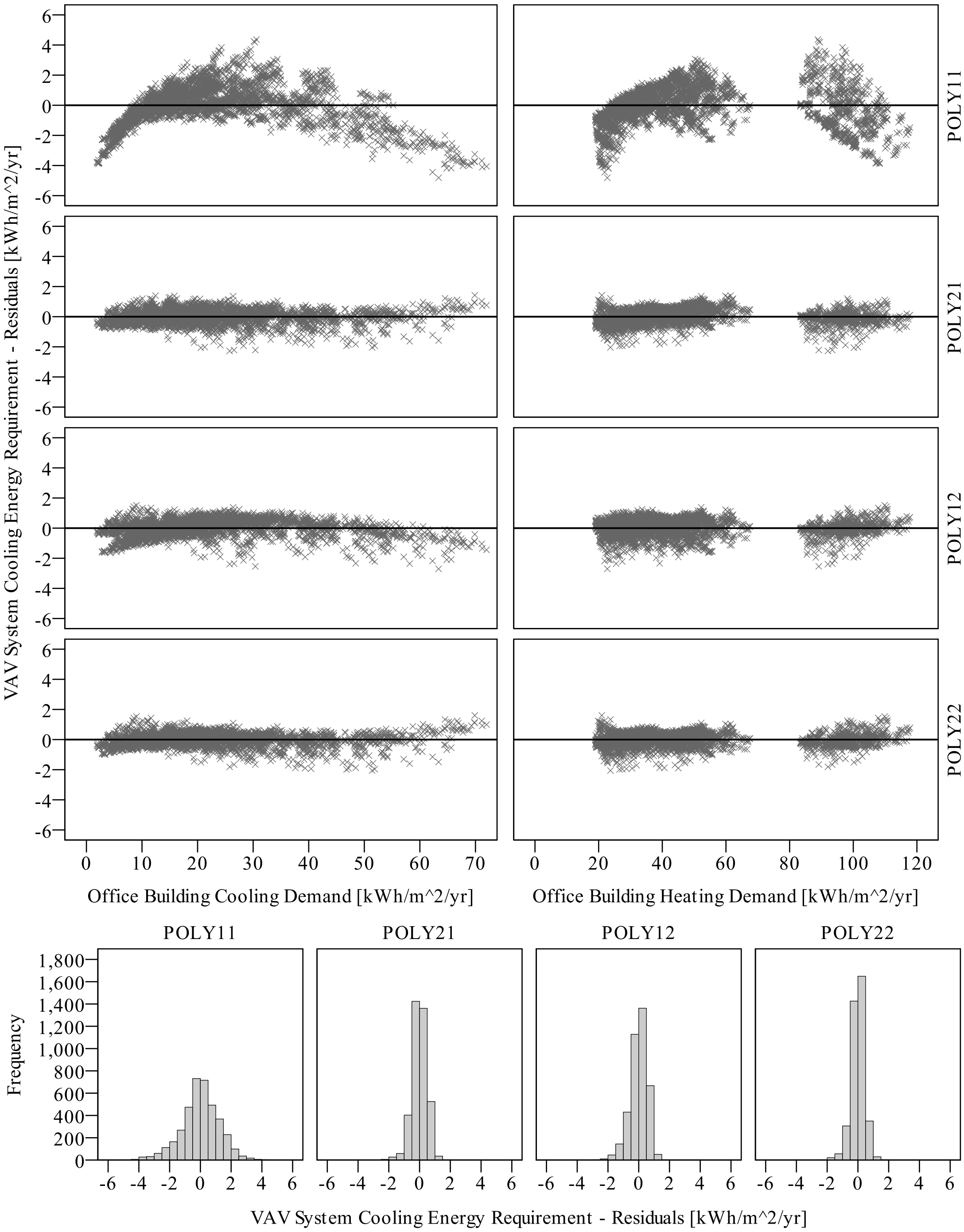
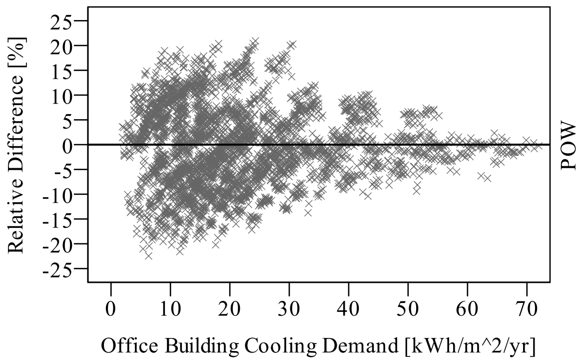


Figure 4.12. Residuals scatter plots and histograms for two independent variables models of VAV system cooling energy consumption in office buildings

Previous analyses were based on the analysis of residuals which were defined as differences between predicted and observed values. The same data can be viewed from a different angle by analysing relative differences which were calculated by dividing residuals with the associated observed values. Relative differences scatter plots and distribution histograms for the POW and POLY22 models, which are the best-fit models in each group of analysed regression models, are presented in Figure 4.13. Scatter plots show the expected pattern of decreasing relative differences with an increment of building cooling demand. However, the zone with relatively high relative differences is much smaller in the case of the POLY22 model. The distribution of relative differences presented in histograms visualise the advantages of the full polynomial two independent variables model POLY22, although the POW model is capable of predicting the VAV system cooling energy requirement in majority of simulated scenarios within a ±20% relative difference range. Table in Figure 4.13 top right corner summarised statistical parameters of relative differences for both models. It can be seen that the POLY22 model predicted the VAV system cooling energy requirements with a ±1.75% relative difference for the half of all simulated buildings. In addition, all predictions fell into the range between -20% and 12.5%. Complete table, which includes relative differences for all other analysed models, can be found in Appendix C (Table C.2).



	POW	POLY22
Mean	-0.63	-0.21
Std. Deviation	7.984	3.089
Maximum	20.93	12.49
Minimum	-22.46	-20.21
Percentile 25	-6.37	-1.75
Percentile 75	5.31	1.72

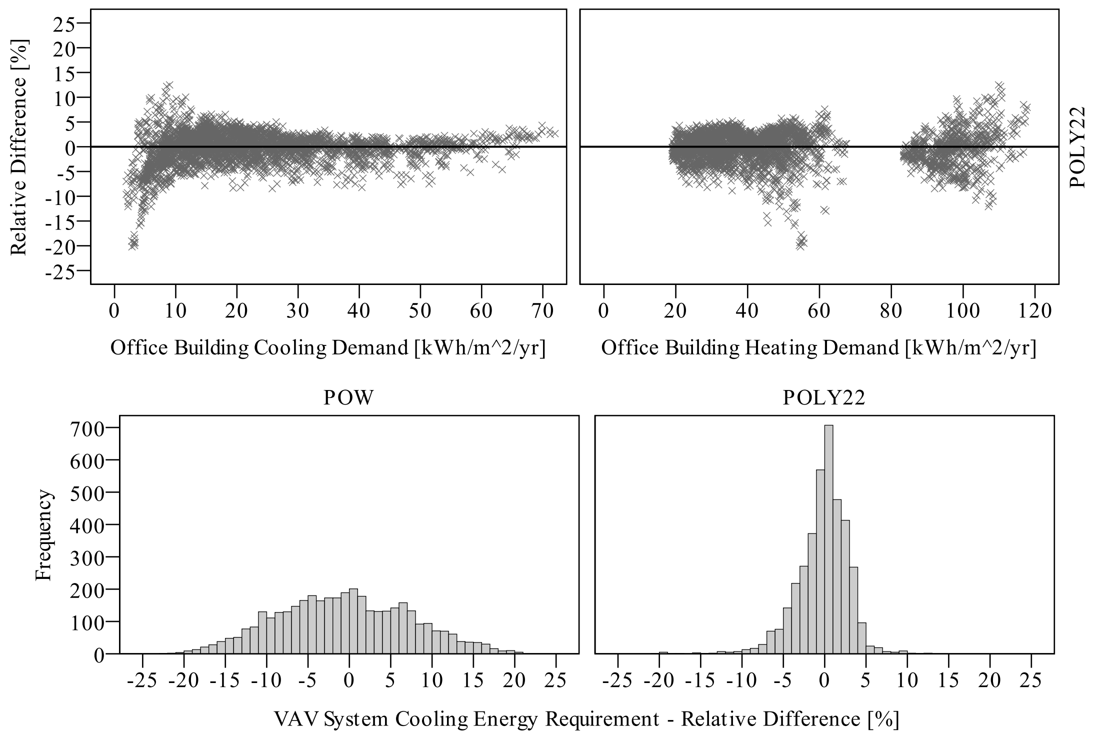


Figure 4.13. Residuals scatter plots and histograms for POW and POLY22 models of VAV system cooling energy consumption in office buildings

Second important parameter in analysis of an HVAC system performance is system heating energy consumption. VAV system heating energy requirement, was also fitted to the same two sets of regression models. The correlation between the VAV system heating energy requirement and the associated building heating demand is presented in Figure 4.14.

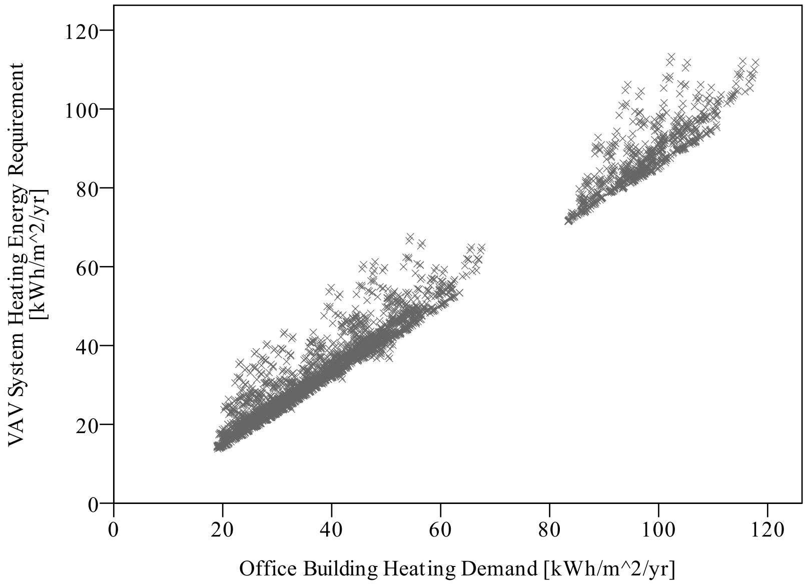


Figure 4.14. VAV system heating energy requirement as a function of office building heating demand

Data in Figure 4.14 indicate almost linear correlation between VAV system heating energy requirements and building heating demands, which has been confirmed by regression analysis results. Fitting the data to the first set of regression models resulted in models best fit parameter values (Table 4.15). Parameter value "c" in the quadratic model (QUAD) is close to zero and the exponent of the power model (POW) is slightly above one which confirms abovementioned high linearity.

Table 4.15. Regression model parameter values for single independent variable models of VAV system heating energy consumption in office buildings

Model	Model Parameters		
	a	b	c
LIN1	-	8.908×10^{-1}	-
LIN2	-1.683	9.175×10^{-1}	-
QUAD	-1.222	8.988×10^{-1}	1.477×10^{-4}
POW	-9.102×10^{-1}	8.442×10^{-1}	1.016

Statistical parameters of single variable models goodness of fit presented in Table 4.16 provide additional information about studied models. The simplest model (LIN1) is less accurate than other models. All other models behave very similar to each other, although a minor improvement can be achieved with both quadratic and power models when compared to predictions of the LIN2 model.

Table 4.16. Comparison of single independent variable models of VAV system heating energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW
$\bar{y}$	43.19	43.57	43.19	43.19	43.19
σ_y	24.415	23.433	24.135	24.135	24.135
RSS		54664	52226	52200	52204
$\mathbf{R^2}$		0.9761	0.9772	0.9772	0.9772
RMSD		3.773	3.6879	3.687	3.6871
e_{max}		22.14	21.31	21.30	21.29
e_{min}		-8.35	8.02	-7.91	-7.92
$\overline{\lvert e \rvert}$		2.70	2.45	2.45	2.45
$\lvert e \rvert_{95\%}$		7.70	7.73	7.81	7.81

Residuals scatter plots and distribution histograms confirm the similarity between all four models (Figure 4.15). The differences between models are hardly visible. One thing which all four models has in common and requires further clarification is the long histogram"s right side tail. This is caused by the characteristic of the system when coupled with different building types. It has been previously mentioned in section 4.1.2 that the building floor space arrangement impacts an air-conditioning system behaviour. This is especially evident in buildings with space arrangement other than open plan, mainly due to zones heating and cooling imbalances. Building type two (cellular daylit building type) is particularly affected by this and most of outliers in scatter plots in Figure 4.15 belong to the VAV system coupled with this particular building type.

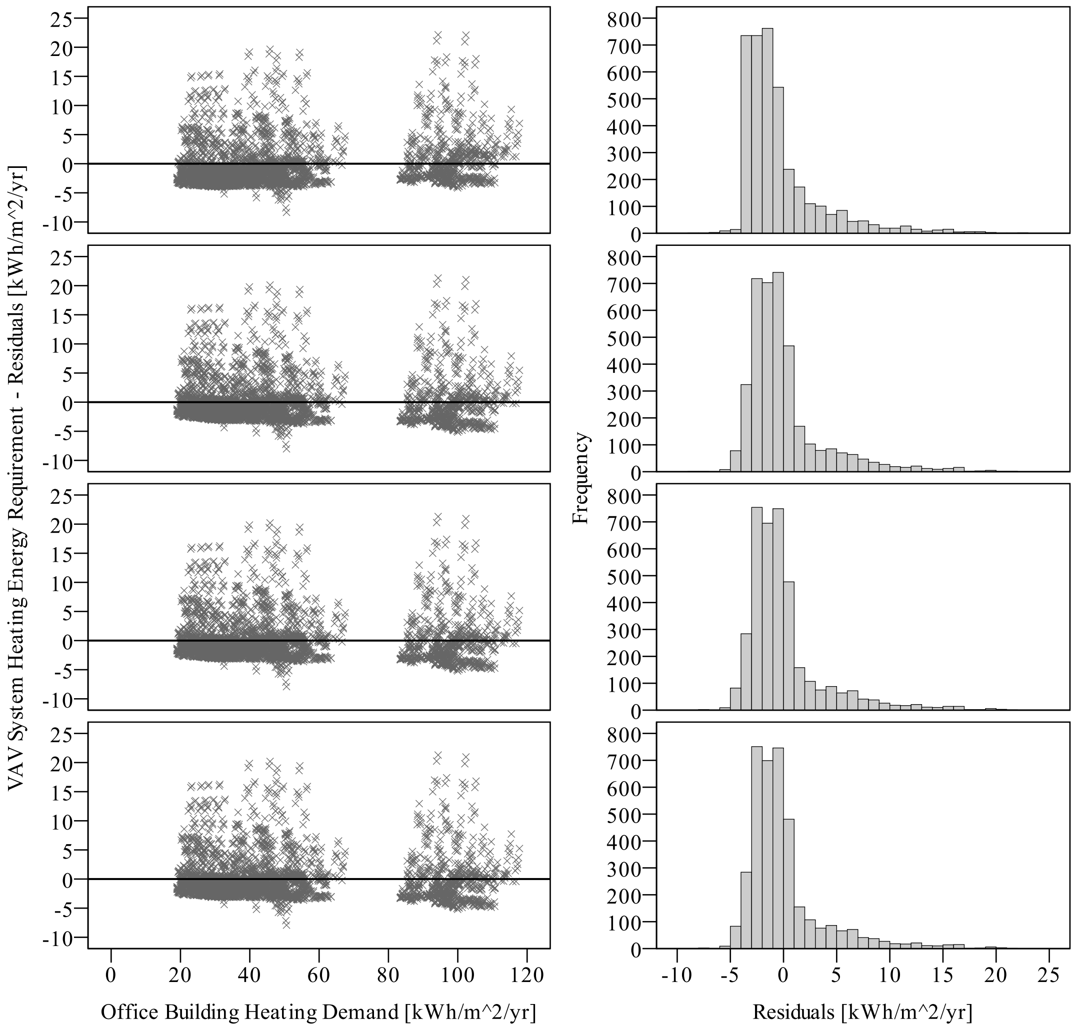


Figure 4.15. Residuals scatter plots and histograms for single independent variable models of VAV system heating energy consumption in office buildings

Fitting the VAV system heating energy requirements to the second set of regression models resulted in determining models best fit parameter values presented in Table 4.17. Standard errors of the best fit parameter values are small and their 95% confidence intervals are narrow, as well as in the case of the best fit parameter values obtained from the first set of regression models. These values are presented in Table C.3 in Appendix C.

Table 4.17. Regression model parameter values for two independent variables models of VAV system heating energy consumption in office buildings

Model	Model Parameters					
	a	b	c	d	e	f
POLY11	-6.516	1.403×10^{-1}	9.545×10^{-1}	-	-	-
POLY21	5.952×10^{-1}	-3.901×10^{-1}	8.535×10^{-1}	4.909×10^{-3}	7.008×10^{-3}	-
POLY12	-3.811	-3.675×10^{-4}	8.693×10^{-1}	-	4.251×10^{-3}	2.560×10^{-4}
POLY22	5.834	-5.225×10^{-1}	6.950×10^{-1}	5.827×10^{-3}	8.896×10^{-3}	1.015×10^{-3}

Statistical parameters of the second set of regression models are presented in Table 4.18. Each of two independent variables models offers improvement in comparison with the previously described single variable models. Coefficient of determination is increased from 0.977 in case of single variable models to a minimum of 0.9815, which is the R^2 of the POLY11 model. Second order models (POLY21, POLY12 and POLY22) further increase the coefficient of determination. The closest fit to observed values was achieved by the POLY22 model which is a full polynomial two independent variables second order model.

Table 4.18. Comparison of two independent variables models of VAV system heating energy consumption in office buildings

	Observed	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	43.19	43.19	43.19	43.19	43.19
σ_y	24.415	24.187	24.218	24.204	24.222
RSS		42440	36754	39306	36072
$\mathbf{R^2}$		0.9815	0.9839	0.9828	0.9842
RMSD		3.3245	3.093	3.1994	3.0649
e_{max}		18.87	15.64	15.83	16.02
e_{min}		-5.94	-7.03	-6.53	-7.35
$\overline{\lvert e \rvert}$		2.42	2.25	2.39	2.16
$\lvert e \rvert_{95\%}$		6.01	6.11	5.85	6.29

Figure 4.16 shows two independent variables regression models residuals scatter plots plotted against both independent variables, as well as residuals distribution histograms. The improvement from single independent models is visible since the distributions are less skewed and the tail is shorter It is also obvious that the POLY22 model outperforms other models as its residuals distribution histogram is the narrower and the maximum bin is the highest when compared to other models.

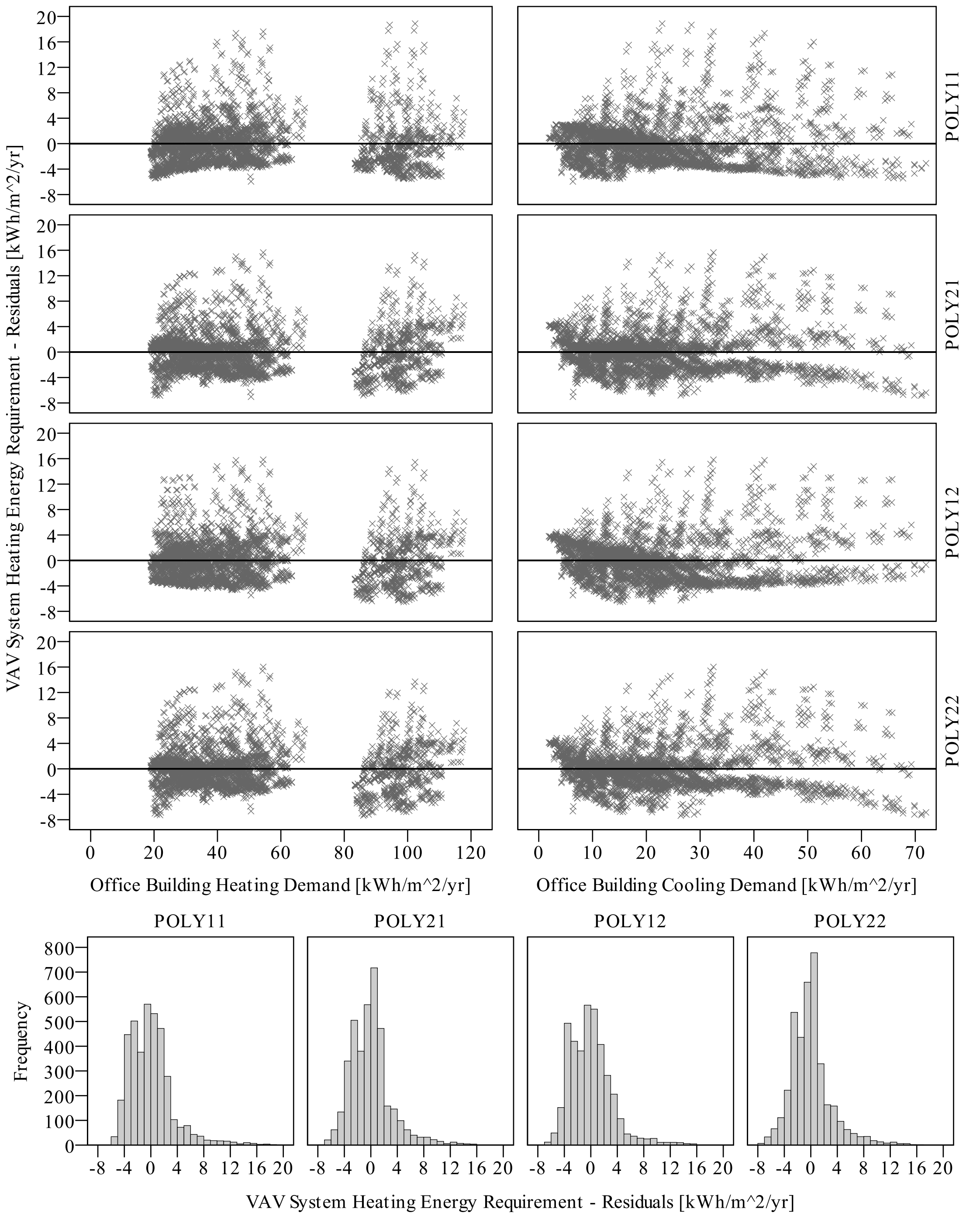
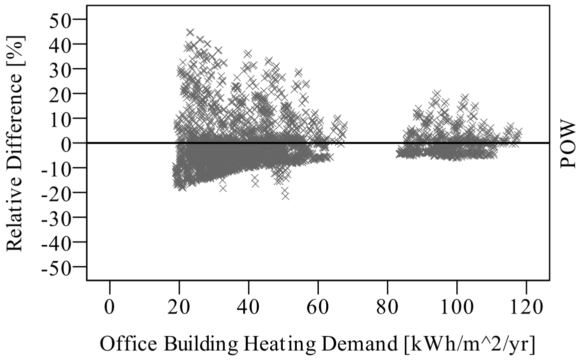


Figure 4.16. Residuals scatter plots and histograms for two independent variables models of VAV system heating energy consumption in office buildings

More information about goodness of fit of regression models can be obtained if relative differences between predicted and observed system heating energy requirements are compared for different models. Figure 4.17 presents relative differences scatter plots and distribution histograms for the power model and the

POLY22 model. Although relative differences distribution histograms of both models look very similar, the POLY22 model is slightly better which can be proved by comparing statistical parameters summarised in table in the Figure 4.17 top right corner. The mean of relative differences is closer to zero in the case of the POLY22 model and the standard deviation is 1% lower. Table C.4 in Appendix C presents the relative differences statistical parameters of all analysed models.



	POW	POLY22
Mean	-0.94	-0.68
Std. Deviation	8.885	7.836
Maximum	44.74	31.13
Minimum	-21.56	-41.47
Percentile 25	-5.99	-5.30
Percentile 75	1.17	2.84

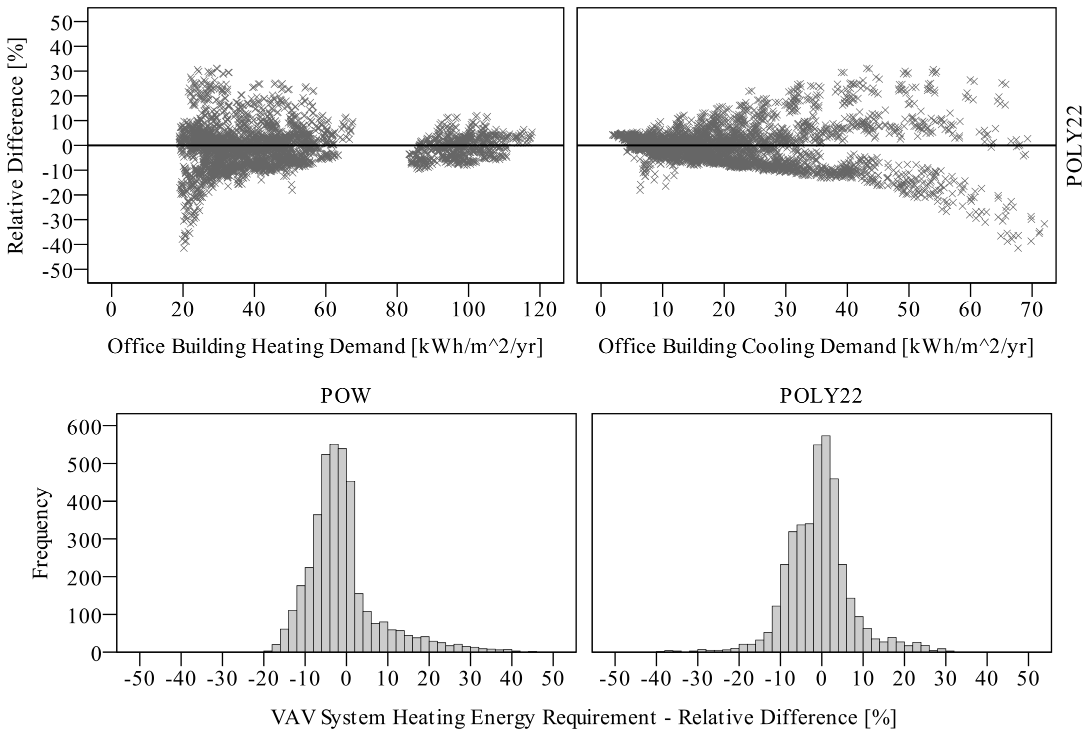


Figure 4.17. Residuals scatter plots and histograms for POW and POLY22 models of VAV system heating energy consumption in office buildings

In addition to the system cooling and heating energy consumptions, regression models were also used to fit the VAV system auxiliary energy requirements data. The relationship between the VAV system auxiliary energy requirements and associated building total demands is presented in Figure 4.18.

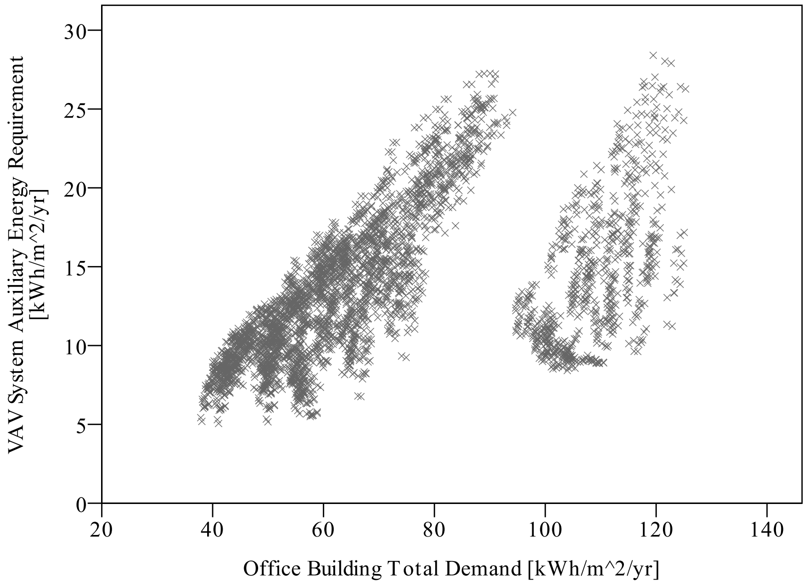


Figure 4.18. VAV system auxiliary energy requirement as a function of office building total demand

Since there is no good correlation between auxiliary energy requirements and building total demands which could allow using any of single independent variable models, the auxiliary energy requirements data were fitted only to the second set of regression models. Regression models best fit parameter values are presented in Table 4.19 while standard errors and 95% confidence bounds are shown in Table C.5 in Appendix C.

Table 4.19. Regression model parameter values for two independent variables models of VAV system auxiliary energy consumption in office buildings

Model	Model Parameters					
	a	b	c	d	e	f
POLY11	3.436×10^{-2}	3.779×10^{-1}	1.087×10^{-1}	-	-	-
POLY21	1.586	2.556×10^{-1}	4.554×10^{-2}	-8.386×10^{-4}	5.409×10^{-3}	-
POLY12	2.709	1.840×10^{-1}	2.979×10^{-2}	-	5.995×10^{-3}	4.549×10^{-5}
POLY22	1.213	2.650×10^{-1}	5.682×10^{-2}	-9.039×10^{-4}	5.274×10^{-3}	-7.225×10^{-5}

Regression models statistical parameters summarised in Table 4.20 show that stepping up from the first order polynomial model POLY11 to the second order polynomial models considerably improves goodness of fit. Coefficient of determination rose from close to 0.875 in the POLY11 model to above 0.96 in all other models. It is also worth to mention that second order polynomial models are very close to each other, particularly there is almost negligible difference between POLY21 and POLY22 models.

Table 4.20. Comparison of two independent variables models of VAV system auxiliary energy consumption in office buildings

	Observed	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	13.50	13.50	13.50	13.50	13.50
σ_y	4.605	4.306	4.518	4.515	4.518
RSS		10217	3066	3141	3063
R^2		0.8745	0.9623	0.9614	0.9624
RMSD		1.6312	0.8936	0.9044	0.8931
e_{max}		7.33	4.00	4.06	3.96
e_{min}		-5.55	-3.90	-4.10	-3.83
$\overline{\lvert e \rvert}$		1.10	0.80	0.69	0.68
$\lvert e \rvert_{95\%}$		3.54	1.83	1.88	1.80

VAV system auxiliary energy requirements residuals scatter plots and distribution histograms presented in Figure 4.19 confirm that the POLY11 model does not provide a good fit since there are several large clustering of adjacent residuals. Scatter plots and distribution histograms of all other models are nearly the same. It is almost impossible to spot any obvious difference.

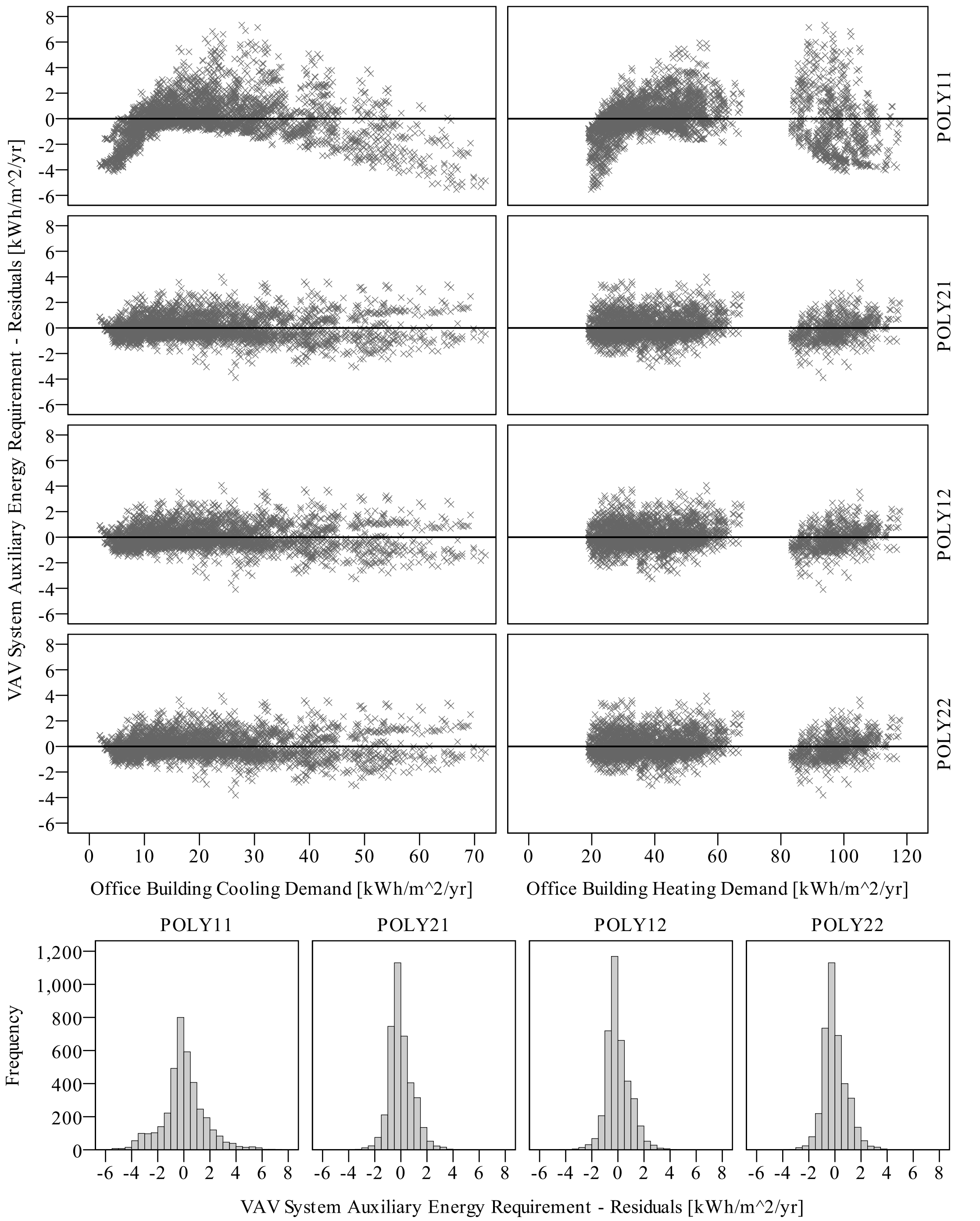
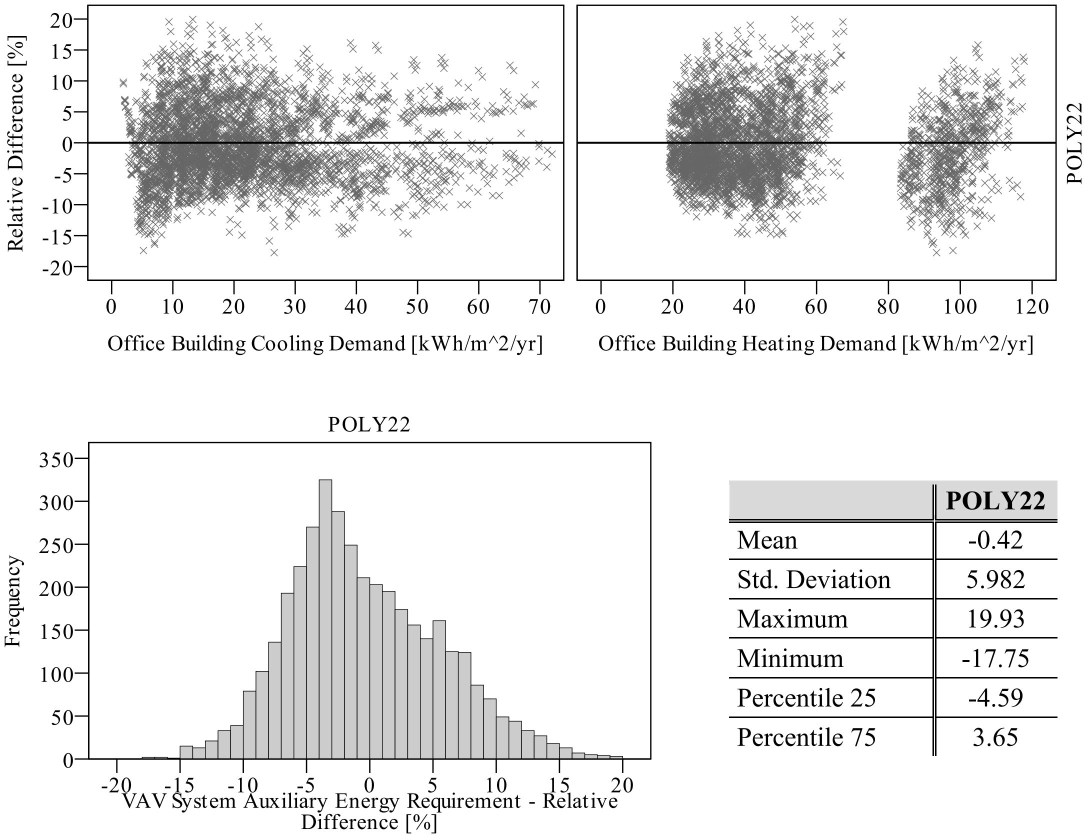


Figure 4.19. Residuals scatter plots and histograms for two independent variables models of VAV system auxiliary energy consumption in office buildings

Since the POLY22 model had the smallest root mean square deviation and the highest coefficient of determination, its auxiliary energy requirement relative differences scatter plots and distribution histogram is presented in Figure 4.20. It can be seen that the POLY22 model is capable of predicting the VAV system auxiliary energy requirements within ±20% range of relative differences. 50% of all predictions fell between -4.6% and 3.65%, which is presented in table of Figure 4.20, bottom right corner. Table C.6 in Appendix C summarised the relative differences statistical parameters of all other regression models used to fit the VAV system auxiliary energy requirements data.



	POLY22
Mean	-0.42
Std. Deviation	5.982
Maximum	19.93
Minimum	-17.75
Percentile 25	-4.59
Percentile 75	3.65

Figure 4.20. Residuals scatter plots and histogram for POLY22 model of VAV system auxiliary energy consumption in office buildings

4.2.2.1. The correlation between HVAC systems energy use and building cooling and heating demand

VAV system analysis showed that models which are a function of both the building heating and cooling demand are more accurate than single-variable models that are a function of only heating or cooling demand. While it is logical to fit HVAC system cooling energy requirements to zone cooling demand and HVAC system heating energy requirements to zone heating demand alone, the decision to fit them to both the heating and cooling demand requires further clarification.

Building zone heating demand and cooling demand were calculated by taking into consideration typical heat gains and heat losses that occur in buildings. This means that building heating and cooling demands capture only the characteristics of a building and its operation, in addition to the impact of a weather and location. The zone demand calculation does not take into account the impact that a particular HVAC system can have on the building total energy requirements. The table of correlation coefficients (Table 4.21) shows that the building heating demand is most greatly affected by building fabrics (correlation coefficient of -0.87), while the building cooling demand is most greatly affected by glazing ratio (0.52), glazing coating (-0.47) and building fabrics (0.5). On the other hand, the correlation coefficients reveal that system cooling energy requirements are mostly affected by building cooling demands (0.98) and system heating energy requirements are mostly affected by building heating demands (0.99), which was expected. They also reveal that the building cooling demand has significant impact on the system heating requirements (-0.46) as well as that the building heating demand has significant impact on the system cooling requirements (-0.41). It can also be seen that the impact of building parameters on system heating and cooling energy requirements is close to their impact on building heating and cooling demand. Findings from the table of correlation coefficients led to the decision to fit HVAC systems cooling and heating energy requirements to both the building heating and cooling demand.

Table 4.21. Building demands and system energy requirements correlations

	Type	**Orientation**	**Glazing %**	**Gl. Coating**	**Overhangs**	**Daylight**	**Build. Fabric**	**Heating Demand**	**Cooling Demand**
Heating Demands	-0.03	0.00	0.09	0.13	0.03	0.12	-0.87		
Cooling Demand	0.00	0.00	0.52	-0.47	-0.18	-0.27	0.50		
VAV Sys Heating	-0.01	-0.01	0.14	0.08	0.02	0.11	-0.86	0.99	-0.46
VAV Sys Cooling	0.01	0.00	0.63	-0.48	-0.20	-0.26	0.38	-0.41	0.98

4.2.2.2. Recommended models – POW and POLY22

VAV system"s cooling and heating energy requirements calculated by EnergyPlus were fitted to two sets of regression models, while the auxiliary energy requirements were fitted to the second set of regression models only. First set of regression models is based on single independent variable (either building cooling demand or building heating demand) and the second set of regression models are function of two independent variables (both building heating and cooling demand). Since the aim of this research is to provide simplified, yet accurate, mathematical models which can predict HVAC system energy requirements as a function of building demands, adopting a single model from each of these two groups of regression models for all analysed systems is a clear advantage. In this way, possible misinterpretation and mistakes are minimized since the user has to choose only model best fit parameter values, which depend on the HVAC system type, and apply them to a simple equation.

The quality of statistical models was measured against root mean square value and coefficient of determination. The best model from the second group of regression models is the POLY22 and is as such recommended. POLY22 model also provides better fit than any of single variable regression models from the analysed set of data. The recommended model from the first set of regression models is the power model although the POW model is not always a model with the best statistical parameters. In some cases the quadratic model provides better fit. However, the goodness of HVAC system energy requirements predictions is not compromised in cases where the quadratic model provides better fit since statistical parameters of both models are very close to each other. This can be seen in Table 4.22. Highlighted fields address the cases where the quadratic model predictions are more accurate than predictions of the power model (60% of all cases). Nevertheless, differences between these two models are minor and in some cases do not even affect the coefficient of determination, for example in cases of VAV and CAV systems heating energy requirement predictions.

Table 4.22. Differences between QUAD and POW models

System		Cooling QUAD	Cooling POW	Heating QUAD	Heating POW
VAV	RSS	6041	5992	52200	52204
	R^2	0.9635	0.9638	0.9772	0.9772
CAV	RSS	6451	6463	46813	46877
	R^2	0.9408	0.9406	0.9762	0.9762
FC	RSS	2806	2803	17068	17204
	R^2	0.9978	0.9978	0.9902	0.9901
EMB	RSS	31031	30909	20324	20523
	R^2	0.9771	0.9772	0.9825	0.9823
ALU	RSS	22069	21340	20946	21164
	R^2	0.9765	0.9773	0.9817	0.9815

The advantage is given to the power model due to fact that it does not have a turning point. Figure 4.21 visualises the potential issue with the quadratic model. Data in the graph represent VAV system cooling energy requirements altogether with the quadratic model and the power model curves based on this data. Both models behave very similar within the data range. However, soon after the maximum building cooling demand, the quadratic model reaches its maximum value after which it starts declining. This behaviour is not realistic for the HVAC systems. Most likely the system requirements would follow the power line. Large extrapolations are however not recommendable since the regression models were generated from a limited data range.

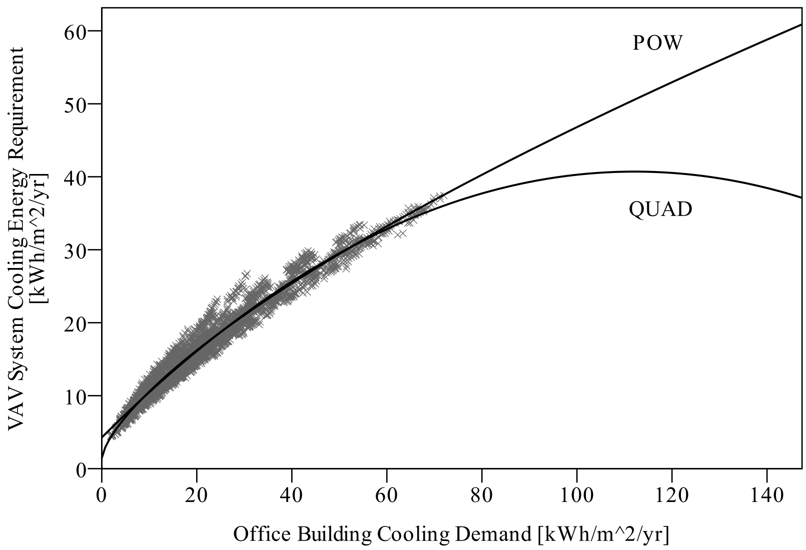


Figure 4.21. VAV system cooling energy requirement – POW and QUAD models

The same analysis employed for the VAV system, was also conducted for all other systems studied. A large number of charts and tables were generated for each individual system and they are presented in Appendix C. A summary of regression analysis for CAV, FC and chilled ceiling systems presented in following chapters includes only models best fit parameter values and associated models statistical parameters of the recommended models (POW and POLY22).

4.2.3. CAV system

The CAV system belongs to the same group of HVAC systems as the VAV system; all-air HVAC systems. Three charts in Figure 4.22 show the correlation between: the CAV system cooling energy requirements and building cooling demands, the CAV system heating energy requirements and building heating demands, as well as the CAV system auxiliary energy requirements and total building demands. It can be seen that the CAV system cooling/heating energy requirements are in strong correlation to the building cooling/heating demands, while the same cannot be said for the auxiliary energy requirements dependence on the total building demands. Shapes of these three scatter plots are very similar to the VAV system scatter plots presented in Figure 4.10, Figure 4.14 and Figure 4.18. The most noticeable difference is in the shape of the heating energy requirement data. Data in the CAV system heating energy requirement scatter plot are more uniform without outliers which exist in the VAV system heating energy requirement scatter plot.

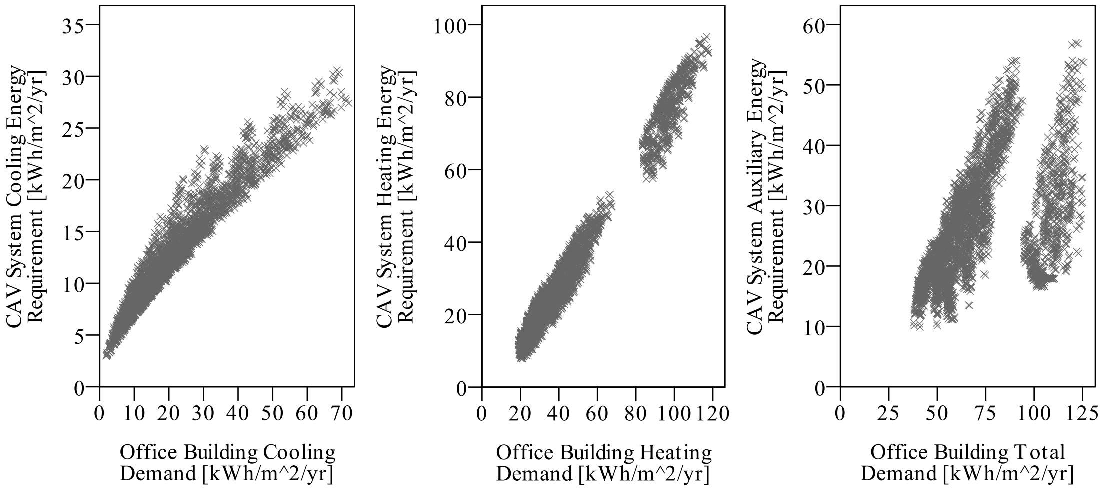


Figure 4.22. CAV system cooling/heating/auxiliary energy requirement as a function of office building cooling/heating/total demand

The CAV system energy requirement data were fitted to all models from the both sets of regression models described previously. Detailed outputs from the regression analysis can be found in Appendix C (Tables C.7 – C.15 and Figures C.1 – C.8).

Table 4.23 summarised the regression analysis outputs for the POW and POLY22 models. Models best fit parameter values are presented in the top part of the table. Exponent c in the heating energy requirement POW model is slightly above one (1.08) which indicates CAV systems‟ almost linear dependence on the building heating demand. This is also noticeable from the figure above. Statistical parameters of regression models are presented in the table‟s bottom part. Overall conclusion from the data presented in the table is that each of recommended models is capable of predicting energy requirements very close to the observed data. The smallest coefficient of determination is 0.94 in the case of the cooling energy requirement power model. On the other hand, the closest fit was obtained by the heating energy requirement POLY22 model which R^2 is 0.997.

Table 4.23. Regression model parameter values for models of CAV system cooling/heating/auxiliary energy consumption in office buildings and models statistical parameters

	Cooling		Heating		Auxiliary
	POW	**POLY22**	**POW**	**POLY22**	**POLY22**
a	7.539×10^{-1}	1.78	-2.625	4.827	2.609
b	1.431	4.144×10^{-1}	5.679×10^{-1}	-2.286×10^{-1}	4.733×10^{-1}
c	7.026×10^{-1}	1.434×10^{-2}	1.08	7.485×10^{-1}	1.097×10^{-1}
d	-	-1.494×10^{-3}	-	2.621×10^{-3}	-1.660×10^{-3}
e	-	2.981×10^{-3}	-	-5.547×10^{-3}	1.138×10^{-2}
f	-	-1.235×10^{-4}	-	7.650×10^{-4}	-1.843×10^{-4}
$\bar{y}$	12.67	12.67	35.72	35.72	26.46
σ_y	5.165	5.269	22.387	22.624	8.875
RSS	6463	2294	46877	5898	15445
R^2	0.9406	0.9789	0.9762	0.997	0.9514
RMSD	1.2973	0.7729	3.4939	1.2393	2.0055
e_{max}	6.57	2.86	6.98	3.64	8.41
e_{min}	-2.57	-3.08	-13.25	-5.40	-7.89
$\overline{\lvert e \rvert}$	0.99	0.59	2.90	0.98	1.55
$\lvert e \rvert_{95\%}$	2.47	1.61	6.25	2.37	4.01

4.2.4. FC system

Fan coil system, which belongs to the air-water HVAC system type, is the system in which the energy between the hot/chilled water and the zone air is exchanged by a forced convection. The system cooling, heating and auxiliary energy requirements as a function of building cooling, heating and total demands respectively is presented in three charts in Figure 4.23. The most notable is the high linearity of the system cooling requirement dependence on the building cooling demand. The split of the heating energy requirement dataset into three parallel groups is also obvious as well as that the auxiliary energy requirements do not correlate to the building total demands.

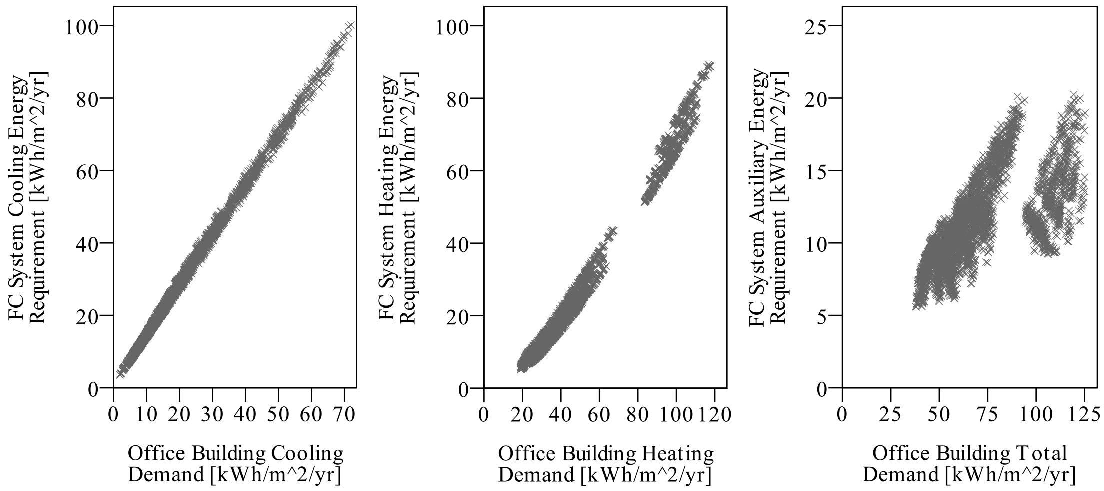


Figure 4.23. FC system cooling/heating/auxiliary energy requirement as a function of office building cooling/heating/total demand

These data were fitted to the same sets of regression models as previous systems. Complete set of outputs from the regression analysis is presented in Appendix C in Tables C.16 – C.24 and Figures C.9 – C.16. The summary for the recommended models is presented in

Table 4.24 which includes the models best fit parameter values and statistical parameters on goodness of fit of these models. Cooling and heating models, both POW and POLY22, have very high coefficient of determination. The lowest one among these four models has R^2 higher than 0.99, while the best one has R^2 above 0.999 which can be accepted as almost perfect fit. Even the auxiliary energy requirements model has the coefficient of determination close to 0.98 which is very high.

One possible reason why FC system cooling and heating models have the coefficient of determination of almost one is the decentralised equipment (four-pipe fan-coil units) used to meet the zone heating/cooling demands. In the all-air systems (VAV and CAV) the energy carrier (air) is pre-treated centrally at a main AHU and then delivered to individual zones where it sometimes needs to be reheated either to cover heating requirements or to prevent overcooling. In decentralised systems such as Fan-coil system, individual zone heating/cooling demands are covered locally by in-zone equipment; in this particular case by four-pipe fan-coil units. In this way the impact of which zones with unequal demands might have on each other, as well as on the HVAC system operation, is minimised.

Table 4.24. Regression model parameter values for models of FC system cooling/heating/auxiliary energy consumption in office buildings and models statistical parameters

	Cooling		Heating		Auxiliary
	POW	POLY22	POW	POLY22	POLY22
a	2.920×10^{-1}	3.376	-1.774	-6.391	2.79
b	1.54	1.435	1.336×10^{-1}	8.287×10^{-3}	1.800×10^{-1}
c	9.732×10^{-1}	-7.328×10^{-2}	1.361	5.476×10^{-1}	4.722×10^{-2}
d	-	-9.210×10^{-4}	-	6.989×10^{-4}	-3.450×10^{-4}
e	-	-1.109×10^{-3}	-	-1.134×10^{-3}	2.598×10^{-3}
f	-	5.069×10^{-4}	-	2.142×10^{-3}	1.052×10^{-4}
$\bar{y}$	30.72	30.72	26.63	26.63	11.34
σ_y	18.304	18.315	21.184	21.188	2.931
RSS	2803	1180	17204	16574	792
R^2	0.9978	0.9991	0.9901	0.9905	0.9765
RMSD	0.8544	0.5543	2.1167	2.0775	0.4541
e_{max}	3.14	1.93	5.66	6.05	1.68
e_{min}	-3.32	-3.31	-4.74	-4.89	-2.66
$\overline{\lvert e \rvert}$	0.64	0.38	1.73	1.70	0.34
$\lvert e \rvert_{95\%}$	1.80	1.24	4.08	4.01	0.87

4.2.5. Chilled ceiling systems – EMB and ALU

In addition to VAV, CAV and FC systems, chilled ceiling systems were also analysed in this research. Chilled ceiling systems belong to the same type of HVAC systems as FC system (the air-water HVAC system type) although they differ from the FC system in the way the energy is exchanged with the zone air. In chilled ceiling systems, most heating and cooling demands are covered by natural convection. Two types of chilled ceiling systems were studied, with the only difference between them in the type of radiant element. The radiant element in the first system is a concrete ceiling with embedded chilled water pipes (EMB system), while the second system uses exposed aluminium panels to maintain desired cooling setpoint (ALU system). The radiator heating covered the heating demand in both systems. Figure 4.24 presents cooling, heating and auxiliary energy requirements correlation on building cooling, heating and total demands respectively for both systems. The only remarkable difference in energy requirement profiles between two analysed systems appears in the cooling energy requirement profile showing that the EMB system has higher cooling demand. Heating and auxiliary energy requirement profiles are almost identical for both systems.

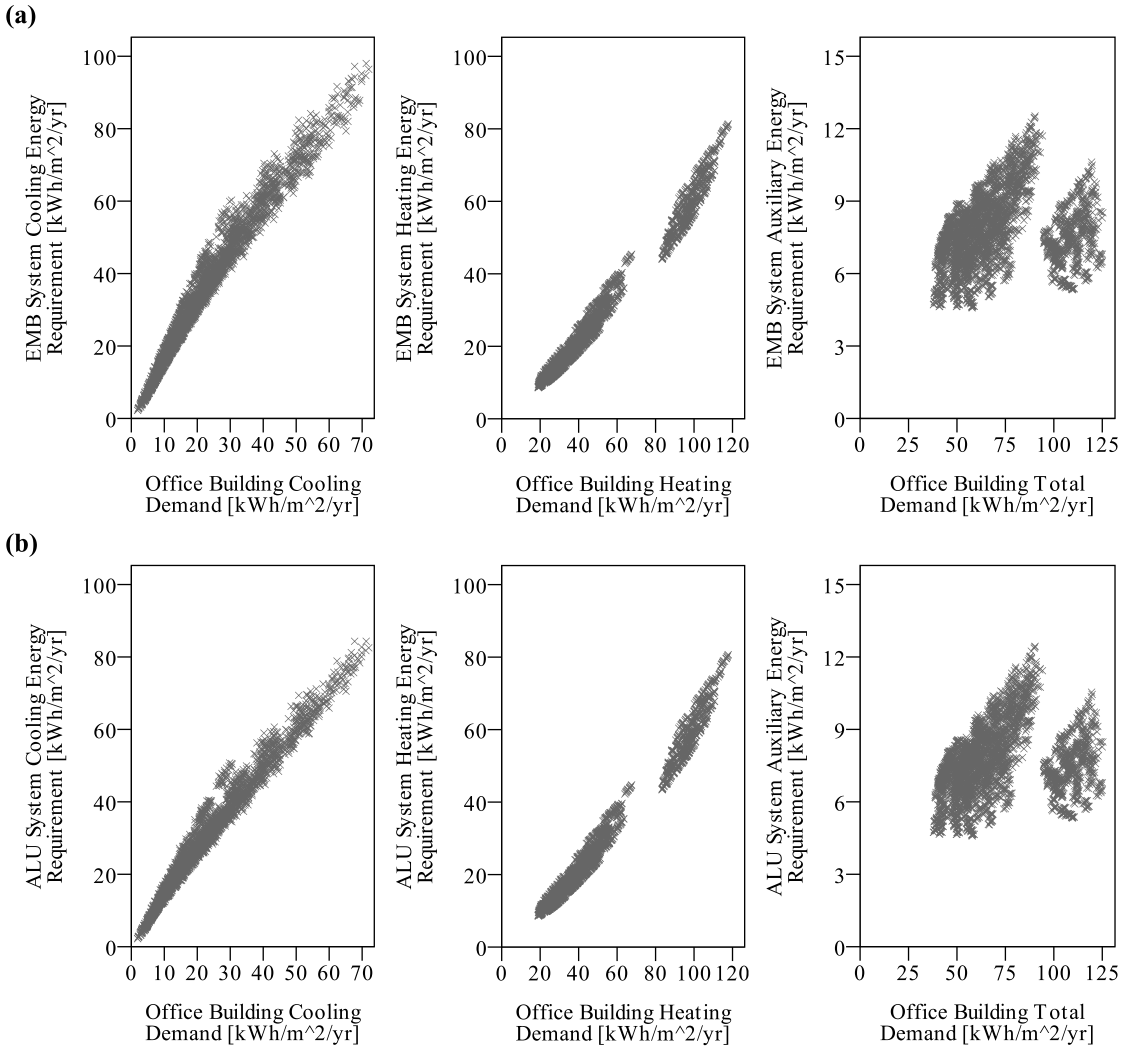


Figure 4.24. Chilled ceiling systems cooling/heating/auxiliary energy requirement as a function of office building cooling/heating/total demand (a: embedded pipes (EMB), b: Aluminium panels (ALU))

Complete chilled ceiling systems regression analysis outputs are presented in Appendix C. Tables C.25 – C.33 and Figures C.17 – C.24 describe the EMB system while the data on the ALU system is stored in Tables C.34 – C.42 and Figures C.25 – C.32. The next table (Table 4.25) summarises the power and POLY22 models best fit parameter values and the associated statistical parameters for both chilled ceiling systems. Parameters of goodness of fit confirm that both POW and POLY22 models can accurately predict heating and cooling energy requirements of both chilled ceiling systems. The coefficient of determination for these models is between 0.977 and 0.992. On the other hand, the R^2 of auxiliary energy requirements model is slightly above 0.87 for both systems, which is the lowest R^2 among recommended models, including all other HVAC systems analysed in this research. Beside the relatively low R^2, the model, which predicts auxiliary energy consumption of chilled ceiling systems, has another drawback. Model predictions are clustered into two subsets. First subset is smaller and includes underestimated cases while the larger subset is much closer to mean but also have a large set of overestimated cases. This can be seen in Figure C.23 and Figure C.31 in Appendix C. Such behaviour was most likely caused by different fresh air requirements that four analysed building types have. Fresh air requirements were calculated according to the number of occupants, which was much smaller in building type 2 (the cellular sidelit building) when compared to other building types, which directly affected the fan energy consumption. The negative aspects of chilled ceilings auxiliary energy requirements POLY22 models are mitigated by the fact that the energy consumption of pumps and fans in chilled ceiling systems is minor in comparison to the building total energy requirements so the errors in predictions do not have large influence on the overall end-use energy consumption predictions.

Table 4.25. Regression model parameter values for models of chilled ceiling systems cooling/heating/auxiliary energy consumption in office buildings and models statistical parameters (embedded pipes (EMB) and aluminium panels (ALU))

	Cooling		Heating		Auxiliary
	POW	POLY22	POW	POLY22	POLY22
EMB System					
a	-8.974	7.126×10^{-1}	-4.637×10^{-1}	-2.391	3.701
b	5.073	1.536	3.218×10^{-1}	-4.703×10^{-2}	1.662×10^{-1}
c	7.055×10^{-1}	-9.533×10^{-2}	1.141	5.389×10^{-1}	1.784×10^{-2}
d	-	-5.689×10^{-3}	-	8.413×10^{-4}	-9.865×10^{-4}
e	-	7.526×10^{-3}	-	1.402×10^{-3}	2.193×10^{-4}
f	-	5.228×10^{-4}	-	8.683×10^{-4}	4.163×10^{-6}
$\bar{y}$	33.67	33.67	27.39	27.39	7.73
σ_y	18.561	18.675	17.248	17.261	1.336
RSS	30909	14612	20523	18835	1000
R^2	0.9772	0.9892	0.9823	0.9838	0.8727
RMSD	2.8371	1.9507	2.3118	2.2147	0.5103
e_{max}	13.23	9.97	7.45	7.36	1.35
e_{min}	-8.43	-9.32	-5.81	-5.47	-1.70
$\overline{\lvert e \rvert}$	2.13	1.41	1.76	1.68	0.42
$\lvert e \rvert_{95\%}$	5.97	4.12	5.20	4.84	0.94
ALU System					
a	-3.53	7.604	1.474×10^{-1}	-2.72	3.761
b	2.997	9.158×10^{-1}	2.717×10^{-1}	-2.119×10^{-2}	1.503×10^{-1}
c	7.820×10^{-1}	-1.974×10^{-1}	1.174	5.248×10^{-1}	1.942×10^{-2}
d	-	-4.809×10^{-4}	-	6.135×10^{-4}	-8.198×10^{-4}
e	-	9.362×10^{-3}	-	1.253×10^{-3}	3.009×10^{-4}
f	-	9.757×10^{-4}	-	9.733×10^{-4}	-1.706×10^{-5}
$\bar{y}$	28.54	28.54	26.98	26.98	7.63
σ_y	15.476	15.589	17.118	17.133	1.291
RSS	21340	7877	21164	19138	918
R^2	0.9773	0.9916	0.9815	0.9833	0.8745
RMSD	2.3574	1.4322	2.3476	2.2325	0.4889
e_{max}	11.72	7.68	7.26	7.20	1.36
e_{min}	-6.49	-6.07	-5.78	-5.44	-1.52
$\overline{\lvert e \rvert}$	1.68	1.09	1.81	1.71	0.40
$\lvert e \rvert_{95\%}$	4.80	2.93	5.23	4.81	0.90

4.2.6. Evaluation of the regression models

In order to evaluate the ability of the regression models to predict HVAC systems cooling, heating and auxiliary energy requirements, an independent dataset was developed. These validation dataset models shared most of their parameters with the original dataset. However, they differ from the original dataset models in that they have no fenestration in some of the walls as shown in Figure 4.25 (walls marked with bold black lines have no fenestration).

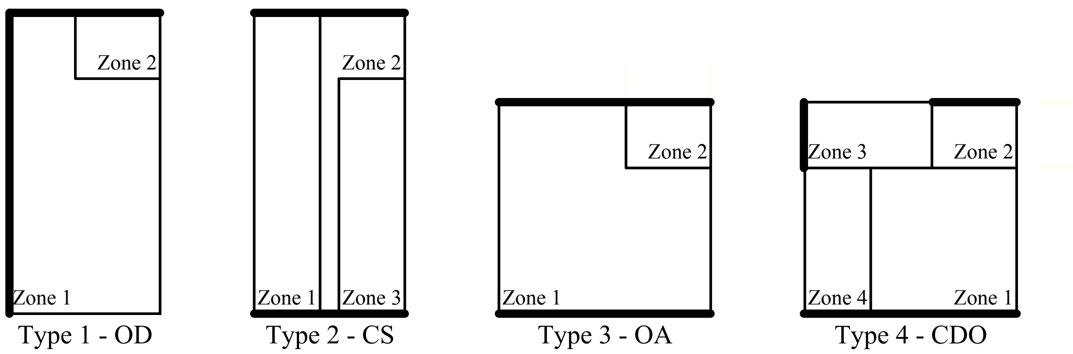


Figure 4.25. Building models used for the validation dataset development

The validation dataset is composed from 1,920 office building models which were coupled with five HVAC systems, as well as with building demand calculations. The simulation outputs were used to evaluate the regression models developed from the original dataset. Statistical parameters which describe the goodness of fit of the regression models applied to both the original and validation dataset are presented in Table 4.26. It can be seen that there are several occasions where the RMSD of the validation dataset is larger than the RMSD of the original dataset. However, since the difference in RMSD"s and residual ranges is not substantial, it can be concluded that the developed regression models, both single-variable and two-variables, are able to predict HVAC systems heating, cooling and auxiliary energy requirements with sufficient accuracy.

Table 4.26. Original/validation dataset regression models statistical parameters

		Cooling				Heating				Auxiliary	
Model		**POW**		**POLY22**		**POW**		**POLY22**		**POLY22**	
Dataset		Original	Validation	Original	Validation	Original	Validation	Original	Validation	Original	Validation
VAV	**RMSD**	1.249	1.389	0.425	0.432	3.687	2.813	3.065	2.558	0.893	1.086
	e_{max}	5.42	3.52	1.60	0.98	21.29	20.25	16.02	16.30	3.96	4.95
	e_{min}	-2.71	-3.77	-2.05	-2.42	-7.92	-5.04	-7.35	-4.95	-3.83	-2.84
	$\overline{\lvert e \rvert}$	0.99	1.20	0.32	0.32	2.45	1.84	2.16	1.56	0.68	0.78
	$\lvert e \rvert_{95\%}$	2.36	2.30	0.85	0.84	7.81	4.80	6.29	5.15	1.80	2.36
CAV	**RMSD**	1.297	1.268	0.773	0.800	3.494	3.448	1.239	1.733	2.006	2.506
	e_{max}	6.57	4.98	2.86	4.09	6.98	6.90	3.64	4.21	8.41	10.89
	e_{min}	-2.57	-2.79	-3.08	-1.84	-13.25	-12.32	-5.40	-6.66	-7.89	-6.34
	$\overline{\lvert e \rvert}$	0.99	1.07	0.59	0.58	2.90	2.92	0.98	1.41	1.55	1.87
	$\lvert e \rvert_{95\%}$	2.47	2.19	1.61	1.60	6.25	5.88	2.37	3.27	4.01	5.24
FC	**RMSD**	0.854	1.241	0.554	0.743	2.117	1.883	2.078	1.849	0.454	0.564
	e_{max}	3.14	5.32	1.93	3.89	5.66	5.89	6.05	6.36	1.68	2.16
	e_{min}	-3.32	-1.49	-3.31	-2.43	-4.74	-3.61	-4.89	-3.53	-2.66	-1.88
	$\overline{\lvert e \rvert}$	0.64	0.87	0.38	0.53	1.73	1.51	1.70	1.47	0.34	0.45
	$\lvert e \rvert_{95\%}$	1.80	2.74	1.24	1.66	4.08	3.76	4.01	3.60	0.87	1.07
EMB	**RMSD**	2.837	2.071	1.951	2.101	2.312	2.318	2.215	2.187	0.510	0.527
	e_{max}	13.23	7.58	9.97	9.78	7.45	4.70	7.36	4.80	1.35	1.43
	e_{min}	-8.43	-9.21	-9.32	-8.29	-5.81	-6.88	-5.47	-6.30	-1.70	-1.74
	$\overline{\lvert e \rvert}$	2.13	1.62	1.41	1.50	1.76	1.73	1.68	1.60	0.42	0.45
	$\lvert e \rvert_{95\%}$	5.97	4.08	4.12	4.38	5.20	5.41	4.84	5.17	0.94	0.90
ALU	**RMSD**	2.357	1.695	1.432	1.871	2.348	2.359	2.233	2.206	0.489	0.501
	e_{max}	11.72	6.95	7.68	8.19	7.26	4.67	7.20	4.79	1.36	1.43
	e_{min}	-6.49	-6.25	-6.07	-4.94	-5.78	-6.82	-5.44	-6.24	-1.52	-1.56
	$\overline{\lvert e \rvert}$	1.68	1.29	1.09	1.39	1.81	1.80	1.71	1.64	0.40	0.43
	$\lvert e \rvert_{95\%}$	4.80	3.43	2.93	4.14	5.23	5.37	4.81	5.11	0.90	0.86

4.3. Summary and limitations

Regression analysis outputs for each HVAC system analysed in this study were presented in this chapter. This section will give a brief summary of major findings from the regression study for all systems. In addition, constraints and limitations of this approach will be addressed and further explored.

4.3.1. Summary

HVAC systems energy consumptions calculated by EnergyPlus for 5 different system types and 3840 different office buildings were fitted to two groups of statistical regression models. The first group contained simple models with a single independent variable. The most suitable model among simple models, which can be used to predict HVAC systems heating/cooling energy requirements as a function of building cooling/heating demands, is based on the power function (equation 4.13)

$$\boldsymbol{y}(\boldsymbol{x}) = \boldsymbol{a} + \boldsymbol{b} \cdot \boldsymbol{x}^{\boldsymbol{c}} \qquad \textbf{4.13}$$

The limitation of this model is that only HVAC systems cooling and heating energy requirements can be predicted. It is not suitable for determining HVAC systems auxiliary energy requirements. Model parameters a, b and c for five studied HVAC systems (variable air volume system, constant air volume system, fan coil system chilled ceiling system with embedded pipes and chilled ceiling system with exposed aluminium panels) are presented in Table 4.27.

Table 4.27. Simple statistical model parameters for predicting HVAC systems cooling and heating energy requirement in office buildings

	Cooling			Heating		
System	**a**	**b**	**c**	**a**	**b**	**c**
VAV	1.449	1.829	6.972×10^{-1}	-9.102×10^{-1}	8.442×10^{-1}	1.016
CAV	7.539×10^{-1}	1.431	7.026×10^{-1}	-2.625	5.679×10^{-1}	1.08
FC	2.920×10^{-1}	1.54	9.732×10^{-1}	-1.774	1.336×10^{-1}	1.361
EMB	-8.974	5.073	7.055×10^{-1}	-4.637×10^{-1}	3.218×10^{-1}	1.141
ALU	-3.53	2.997	7.820×10^{-1}	1.474×10^{-1}	2.717×10^{-1}	1.174

The independent variable in the simple model is either building cooling demand in case of the cooling energy requirement model or building heating demand for the heating energy requirement model. Heating and cooling energy requirement models were developed for the range of building heating and cooling demands between 20 kWh/m^2/yr and 120 kWh/m^2/yr, and 5 kWh/m^2/yr and 75 kWh/m^2/yr respectively. Cooling and heating energy requirement simple model predictions for each of five analysed HVAC systems are presented in Figure 4.26. Each of these lines can replace 3840 EnergyPlus simulation outputs with a relatively high accuracy which was confirmed by coefficients of determination presented in top left corners. These charts confirm that HVAC system has large impact on a building energy end-use. Heating energy requirement chart also shows that the EMB system and the ALU system model curves are overlapping, which was expected since both systems use radiators for heating.

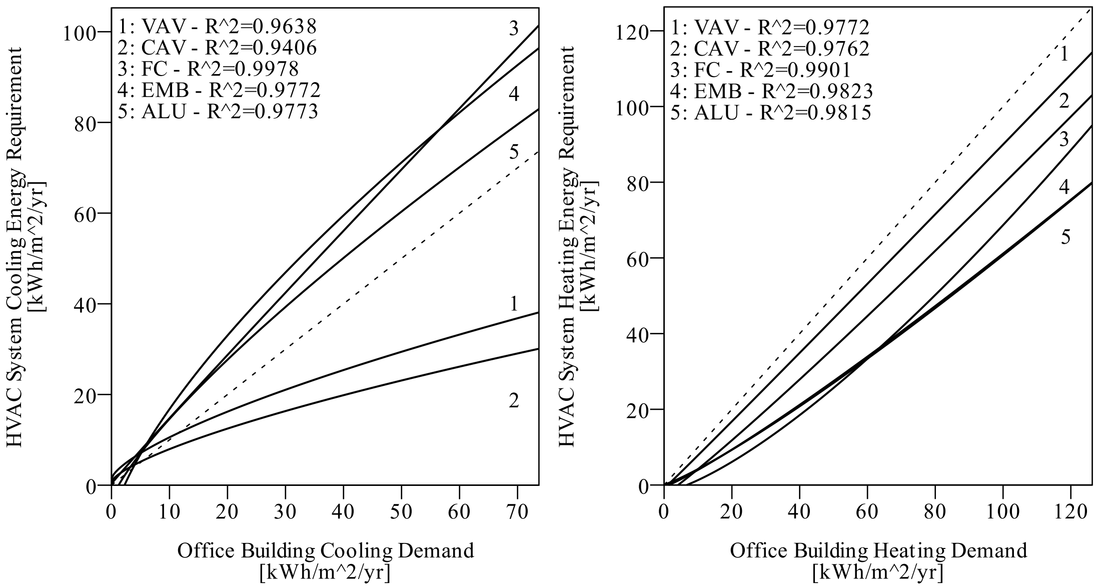


Figure 4.26. HVAC systems cooling and heating simple models predictions

Second group of regression models that HVAC systems were fitted to consists of more complex models based on two independent variables (both building cooling and heating demand). The recommended complex model, which should be used to predict HVAC systems cooling, heating and auxiliary energy requirements, is based on the full second order degree two polynomial function (equation 4.14) where the independent variables x_1 and x_2 are building cooling demand and building heating demand respectively.

$$\boldsymbol{y(x_1, x_2) = a + b \cdot x_1 + c \cdot x_2 + d \cdot x_1^2 + e \cdot x_1 \cdot x_2 + f \cdot x_2^2} \qquad \textbf{4.14}$$

HVAC systems cooling, heating and auxiliary energy requirements complex model parameters a, b, c, d, e and f are presented in Table 4.28.

Table 4.28. Complex statistical model parameters for predicting HVAC systems cooling, heating and auxiliary energy requirement in office buildings

System		Model Parameters					
		a	b	c	d	e	f
Cooling	VAV	1.069	5.792×10^{-1}	7.421×10^{-2}	-2.299×10^{-3}	2.628×10^{-3}	-5.091×10^{-4}
	CAV	1.78	4.144×10^{-1}	1.434×10^{-2}	-1.494×10^{-3}	2.981×10^{-3}	-1.235×10^{-4}
	FC	3.376	1.435	-7.328×10^{-2}	-9.210×10^{-4}	-1.109×10^{-3}	5.069×10^{-4}
	EMB	7.126×10^{-1}	1.536	-9.533×10^{-2}	-5.689×10^{-3}	7.526×10^{-3}	5.228×10^{-4}
	ALU	7.604	9.158×10^{-1}	-1.974×10^{-1}	-4.809×10^{-4}	9.362×10^{-3}	9.757×10^{-4}
Heating	VAV	5.834	-5.225×10^{-1}	6.950×10^{-1}	5.827×10^{-3}	8.896×10^{-3}	1.015×10^{-3}
	CAV	4.827	-2.286×10^{-1}	7.485×10^{-1}	2.621×10^{-3}	-5.547×10^{-3}	7.650×10^{-4}
	FC	-6.391	8.287×10^{-3}	5.476×10^{-1}	6.989×10^{-4}	-1.134×10^{-3}	2.142×10^{-3}
	EMB	-2.391	-4.703×10^{-2}	5.389×10^{-1}	8.413×10^{-4}	1.402×10^{-3}	8.683×10^{-4}
	ALU	-2.72	-2.119×10^{-2}	5.248×10^{-1}	6.135×10^{-4}	1.253×10^{-3}	9.733×10^{-4}
Auxiliary	VAV	1.213	2.650×10^{-1}	5.682×10^{-2}	-9.039×10^{-4}	5.274×10^{-3}	-7.225×10^{-5}
	CAV	2.609	4.733×10^{-1}	1.097×10^{-1}	-1.660×10^{-3}	1.138×10^{-2}	-1.843×10^{-4}
	FC	2.79	1.800×10^{-1}	4.722×10^{-2}	-3.450×10^{-4}	2.598×10^{-3}	1.052×10^{-4}
	EMB	3.701	1.662×10^{-1}	1.784×10^{-2}	-9.865×10^{-4}	2.193×10^{-4}	4.163×10^{-6}
	ALU	3.761	1.503×10^{-1}	1.942×10^{-2}	-8.198×10^{-4}	3.009×10^{-4}	-1.706×10^{-5}

The goodness of fit of HVAC systems energy requirements simple and complex models is presented in Table 4.29. According to coefficients of determination, the complex model can always provide better overall fit. It can be seen from the table that all five HVAC systems cooling and heating models, both simple and complex, have very high coefficient of determination. The lowest R^2 among cooling models is the CAV system, which R^2 is above 0.94 in case of the simple model and close to 0.979 for the complex model. All other systems have higher R^2, which is above 0.99 for the most of complex models. The heating models are also very good, having the R^2 above 0.976 and 0.983 for simple and complex models respectively. In most cases, the complex model offers notable improvement except in both chilled ceiling systems (radiator „wet" heating system) heating models, where the R^2 increment is below 0.002, and in the case of the FC system. The FC system is specific since its R^2 is above 0.99, even for the simple models, which means that there is not too much space for large improvements. Auxiliary energy requirements of VAV, CAV and FC systems can be predicted relatively well with the complex model where the R^2 ranges between 0.95 and 0.98. Not as good performance can be expected in cases of EMB and ALU systems for which the complex model has the coefficient of determination as low as 0.87.

Table 4.29. HVAC systems simple and complex models coefficients of determination

	Cooling		Heating		Auxiliary
System	**Simple**	**Complex**	**Simple**	**Complex**	**Complex**
VAV	0.9638	0.9958	0.9772	0.9842	0.9624
CAV	0.9406	0.9789	0.9762	0.997	0.9514
FC	0.9978	0.9991	0.9901	0.9905	0.9765
EMB	0.9772	0.9892	0.9823	0.9838	0.8727
ALU	0.9773	0.9916	0.9815	0.9833	0.8745

The values of the coefficient of determination presented in Table 4.26 are suggesting that complex model is better for all the systems. However as the quality of the simple statistical model is also very high, the simple mode was recommended as it offers some advantages over the complex model. First of all, the simple model depends only on the one independent variable. For example, if someone needs to calculate only heating energy requirements of HVAC system, the only input which would be required is the building heating demand. Building cooling demand is not needed, which simplifies calculation procedure, as opposed to the complex model for which both demands are obligatory. A second reason lies in the dataset that both models were developed from. The complex model, which depends on both building cooling and heating demand, is not developed to predict HVAC systems energy requirements if both demands are either high or low. If this is the case, the chance to have rather unexpected outputs from the complex model is fairly high, while the simple model has no such limitation. Figure 4.27 visualises the combinations of buildings heating and cooling demands that formed the data set for the development of statistical models. The empty areas in Figure 4.27 thus visualises the zones in which there are no data and determining energy requirements of the building which has, for example, heating demand of 100 kWh/m^2/yr and cooling demand of 70 kWh/m^2/yr with a complex model can lead to the wrong result. Luckily, such buildings are uncommon in the UK climate. Most office buildings can either have a high cooling demand and a low heating demand or vice versa.

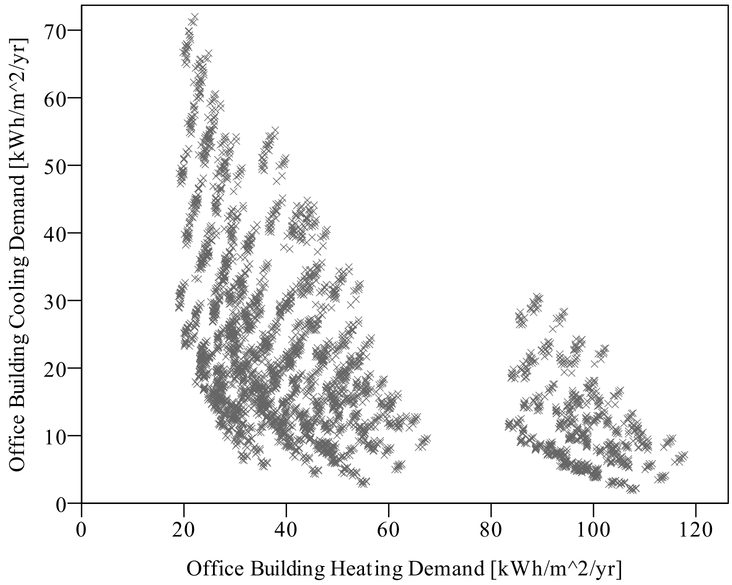


Figure 4.27. Building heating demand vs. building cooling demand

High coefficients of determination which characterise almost all models is a measure of models overall goodness of fit. Analysis of relative differences between models predicted and observed values provides additional information about models performances, although this approach is not perfect since small absolute difference in a case with a low energy requirement can result in a large relative error. For the purpose of the relative differences analysis, percentages of dataset which fall into ±5%, ±10% and ±20% relative difference categories were calculated. These three percentages brackets were chosen to represent excellent fit (±5%), very good fit (±10%) and acceptable fit (±20%).

Table 4.30 shows percentages of five HVAC systems dataset which fell into abovementioned three categories for cooling and heating simple models. It can be seen that the simple model predicted FC system cooling energy requirement within ±5% relative difference in more than 90% cases, which is tremendous especially when taking into account that the second best cooling predictions had only ~50% of data which fell into the excellent fit category (ALU system). The EMB system and the VAV system had 42-43% of data in this category, while the worst performer was the CAV system with only 35% of excellent fit data. The CAV system remained the worst performer in the very good category too, but now with above 65% of all data with relative difference of ±10%, which is a large improvement. Between 73% and 79% of all data are in this category in cases of EMB, ALU and VAV systems. The simple model predicted the FC system cooling energy requirement for almost all cases within the ±10% relative error. Acceptable fit (±20%) was obtained for the complete set of buildings equipped with the FC system and all other systems had the acceptable fit for more than 96.8% of simulated scenarios.

The simple heating model generated slightly less accurate results when compared to the simple cooling model. Only the VAV system had more than 50% of all data in the excellent fit category. All other systems had only between 30% and 40% of predictions in this category. Between 55% and 80% of all data, which depends on the system type, is in the very good fit category, with the CAV system at the lower end and the VAV system at the higher end. The simple model was capable to predict heating energy requirement with acceptable relative error in more than 96% of all data for all systems (including 100% in the case of EMB systems) except for the CAV system which had almost 14% of data outside the ±20% relative difference range. Such a large number of outliers make the simple model less useful in predicting the CAV system heating energy requirements and it has to be used with caution.

Table 4.30. Percentages of simple model predicted values which have relative difference within ±5%, ±10% and ±20% ranges

	Cooling			Heating		
System	**≤ |5%|**	**≤ |10%|**	**≤ |20%|**	**≤ |5%|**	**≤ |10%|**	**≤ |20%|**
VAV	43.07	76.93	99.69	53.59	80.05	95.94
CAV	35.76	65.81	96.80	30.91	56.69	86.51
FC	90.89	99.82	100.00	32.92	63.67	98.05
EMB	41.98	73.02	97.63	39.19	73.80	100.00
ALU	49.66	78.52	98.59	37.89	70.91	99.97

Percentages of HVAC systems cooling, heating and auxiliary energy requirements predictions calculated by the complex model, which fell into ±5%, ±10% and ±20% relative differences categories, are presented in Table 4.31. It is obvious that percentages in these categories are considerably higher for almost all systems in comparison with the simple model, which proves that the complex model is much more accurate. The complex model predicted both cooling and heating energy requirements with ±20% relative difference for more than 99% of data for all systems except in the case of the VAV system for which close to 97% of data are in this category. Very good cooling energy requirement predictions were obtained for more than 90% data in all systems including 100% and 99% for the FC system and the VAV system respectively. Excellent cooling energy requirement fit is obtained in 58.5% in the CAV system, about 65% in chilled ceiling systems, 92% in the VAV system and close to 99% in the FC system.

The complex model heating energy requirement predictions were less accurate when compared to the cooling energy requirement predictions for all models except for the CAV system, according to the relative differences analysis. Heating energy requirements were predicted with ±5% relative difference for close to 40% of all chilled ceiling systems data, 58% of the VAV system data and 75% of the CAV system data. These values are much higher for the very good fit category and amount 80%, 85% and 95% for these systems respectively. The FC system is the only systems for which the complex model does not provide major improvement when compared to the simple model outputs.

HVAC systems auxiliary energy requirements can also be predicted with a high accuracy by using the complex model. Three of five HVAC systems (VAV, FC and ALU) had all data in the ±20% relative difference category and the other two systems had more than 99% of data within this range. Close to 80% of chilled ceiling systems data are in the ±10% category which is the lowest percentage. All other systems had between 86% and 96% of data in this category. Between 48% and 58% of all data are in the best fit category except for the FC system which had more than 80% of data within ±5% relative difference.

Table 4.31. Percentages of complex model predicted values which have relative difference within ±5%, ±10% and ±20% ranges

	Cooling			Heating			Auxiliary		
System	**≤ |5%|**	**≤ |10%|**	**≤ |20%|**	**≤ |5%|**	**≤ |10%|**	**≤ |20%|**	**≤ |5%|**	**≤ |10%|**	**≤ |20%|**
VAV	92.01	99.01	99.97	58.15	85.36	96.80	57.58	91.43	100.00
CAV	58.49	92.40	100.00	75.60	95.55	99.95	48.7	86.67	99.84
FC	98.70	100.00	100.00	32.89	65.70	99.38	80.34	96.30	100.00
EMB	64.30	95.44	99.38	39.95	81.77	100.00	51.20	78.91	99.95
ALU	64.87	94.24	99.19	38.13	78.44	100.00	52.58	79.82	100.00

4.3.2. Limitations

Coefficients of determination and the relative differences analysis showed that both simple and complex models can predict energy requirements of different HVAC systems coupled with office buildings with a great accuracy by knowing only building cooling and heating demands. This conclusion has foundation in the comprehensive dataset the research was based on. However, there are limitations based on the following constraints:

- Climate constraints,
- Office building dataset constraints, and
- HVAC systems constraints.

Climate constraints are related to the weather data used in EnergyPlus simulations. The London Gatwick weather file was used to generate building demands and HVAC systems energy requirements. Building demands and HVAC system energy requirements were regression analysis inputs, which means that recommended models (model parameter values) can be used to determine energy consumption of office buildings located only in the Greater London Area or South East England. They cannot predict energy consumption of buildings located somewhere else since the weather conditions might have significant impact on the HVAC system performance.

In order to examine the influence of different climates on the HVAC systems energy requirements, two additional weather files, Finningley and Aberdeen, were applied to a subset of office buildings models. VAV, CAV and FC systems cooling and heating energy requirements, as well as office buildings cooling and heating demands, were calculated. Systems energy requirements as a function of associated building demands, grouped by a weather condition, are presented in Figure 4.28. It is obvious that building cooling demand decreases and building heating demand increases gradually by changing the weather file form London Gatwick to Finningley and Aberdeen. Such outputs were expected. The interesting point is the response of HVAC systems to the building demands. It can be seen that the systems performance, in particular all-air systems cooling performance, is strongly affected by climate conditions. For example, a building with a cooling demand of 30 kWh/m^2/yr equipped with a VAV system would require 10 kWh/m^2/yr of cooling energy if it is located in Aberdeen, about 17 kWh/m^2/yr if it is in Finningley, and above 20 kWh/m^2/yr if it is placed in London. Such behaviour makes it impossible to create a simple model, with a cooling demand as single independent variable, which can capture different weather conditions. It would be of course possible to develop a simple model for any weather file. This would result in several sets of model parameter values. Each set would be appropriate for particular climate. On the other hand, HVAC system heating energy requirements show a strong correlation with building heating demands almost independent to the UK climate variations. This finding is interesting because it allows a development of a single simple model for the whole UK which can be used to predict particular HVAC system heating energy requirements as a function of building heating demand only.

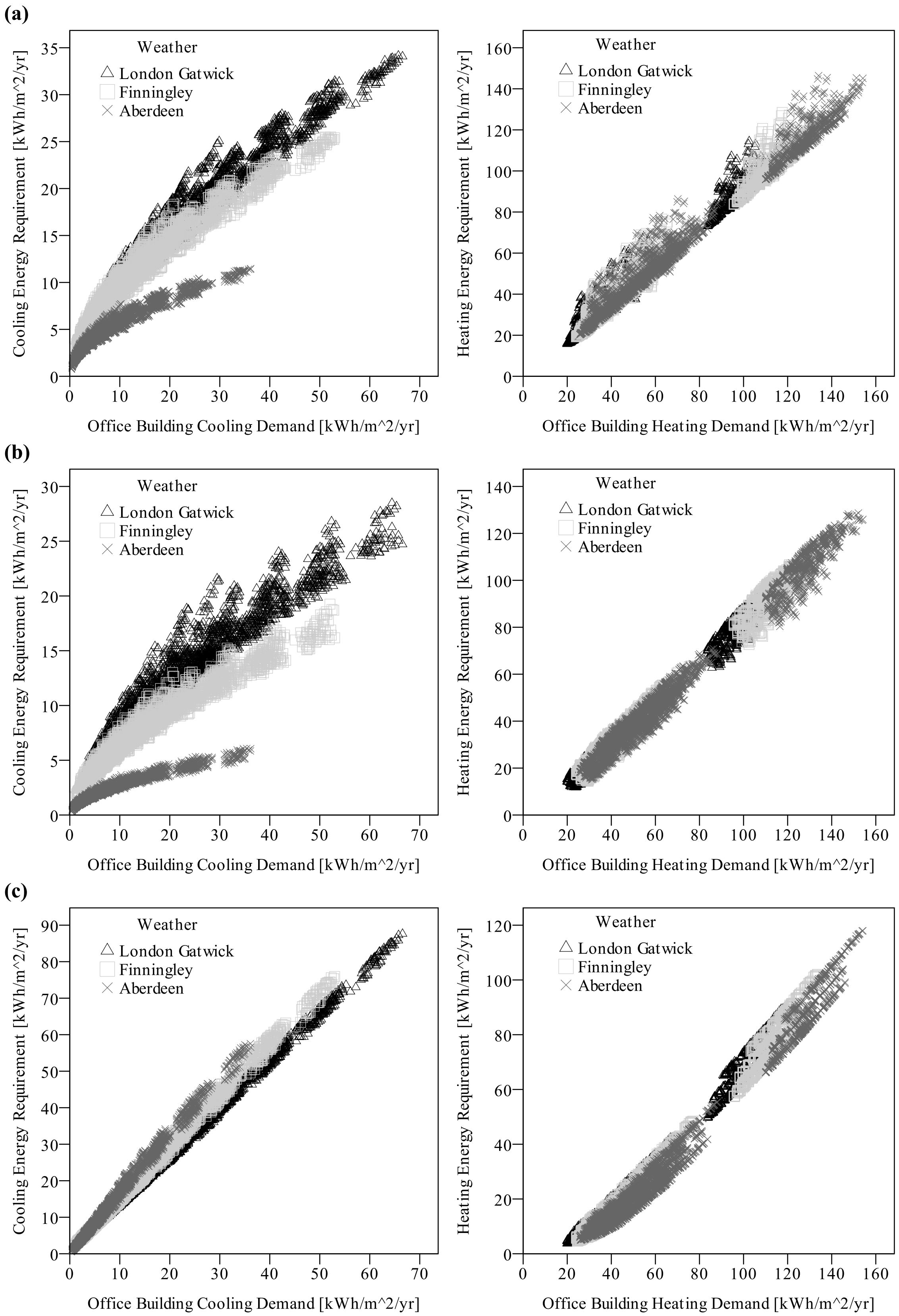


Figure 4.28. VAV (a), CAV (b) and FC (c) system cooling/heating energy requirements vs. building cooling/heating demands

Second set of limitations is related to office buildings models dataset used in this research. Although a large number of building parameters were varied, some of building related parameters, which might have an influence on building energy requirements, were kept fixed, in particular buildings length and depth and form. The dataset could also be extended with buildings of more complex form, beside simplified rectangular and square buildings form shape. Buildings construction elements were adjusted to the UK climate by using materials such as brick and concrete, which means that building models have high thermal mass. Thermally light buildings were not included in the dataset. One more parameter which could be varied, but it was fixed, is the building air tightness. Both very tight and 'leaky' buildings were not analysed.

The last set of constraints, which might have impact on the developed models, includes HVAC systems related parameters. Namely, five HVAC system types coupled with office building models were developed based on standardised assumptions about systems characteristics and control features. Changes in systems parameters might have larger or minor impact on theirs energy requirements, for example a turn-down ratio or a supply air temperature setpoint in a VAV system or heat recovery unit effectiveness in systems with dedicated air. Systems water side and air side resistance as well as pumps and fans efficiencies, which were selected according to various guides and kept constant, directly affect auxiliary energy requirements and indirectly influence cooling and heating energy requirements. Some advanced features, which can be easily implemented in analysed systems and might affect systems requirements, such as night time ventilation, were not taken into account. Chilled ceiling systems are particularly suitable for implementation of advanced techniques, for example coupling with displacement ventilation, or operative temperature control, or even coupling with thermal comfort models. These are all constraints which will affect the accuracy of the HVAC systems energy consumption models developed and presented in this research.

Chapter 5

Conclusions and Further Work

This thesis described the development of regression models which are able to predict, with a high level of accuracy, office building annual heating, cooling and auxiliary energy requirements for different HVAC systems as a function of office building heating and cooling demands. In addition, the developed regression models are shown to be an appropriate tool for selection of a suitable HVAC system for an office building. Building heating and cooling demands were chosen as input parameters as they are relatively easy to calculate at various stages of building design or refurbishment project stages.

In order to represent the office building stock as accurately as possible, a large number of building parameters and their influence on a building energy demand were explored in this study. Chapter two presented the literature review on an office building built form, building construction parameters such as insulation levels, glazing percentages, infiltration, etc. and building parameters related to environmental conditions and building activity which include lighting, appliances and occupancy density. For the purpose of this research, many of these parameters were varied while some of them were kept constant. Four building built forms were coupled with five building fabrics and three levels of glazing. Building orientation was also varied in 45 degree intervals. In addition, two measures of reducing solar gains, overhangs and reflective coating, were considered as well as implementation of daylight control. Selected built forms, insulation levels, glazing percentages, etc. were combined into a large set of office building models (3,840 in total).

Building demand calculation is useful in studies where the main task is to investigate the performance of a building and its components. However, building demands do not give information about a building energy consumption which is largely affected by the choice of HVAC system type. Different HVAC systems have different energy requirements when responding to same building heating and cooling demands. Chapter three presented the classification of HVAC systems and briefly described basic characteristic of various HVAC system types as well as the literature review on the energy efficiency of different HVAC systems. Chapter three also described, in more detail, five HVAC system models, developed for the purpose of this research, which were coupled with previously described office building models. These five systems are: variable air volume system (VAV), constant air volume system (CAV), fan coil system with dedicated air (FC), chilled ceiling system with embedded pipes, dedicated air and radiator heating (EMB), and chilled ceiling system with exposed aluminium panels, dedicated air and radiator heating (ALU).

Annual heating and cooling demands as well as heating, cooling and auxiliary energy requirements of five selected HVAC systems were calculated for each of 3840 office building models by using EnergyPlus building simulation software. The EnergyPlus outputs were normalised per meter square and stored in the large database. The analysis of the simulation outputs were conducted in two steps:

- Analysis of office buildings energy requirements when coupled with different HVAC systems.
- Development of mathematical models which predict HVAC systems heating, cooling and auxiliary energy requirements as a function of building heating and cooling demands.

In the first part of the analysis, cooling, heating and auxiliary energy requirements of different HVAC systems coupled with office building stock were compared among themselves as well as with the building demands. Major conclusions were drawn from the analysis of statistical parameters of each of these database subsets, such as mean, standard deviation, range, etc. It was found that office buildings equipped with all-air systems (VAV and CAV) in general have lower cooling energy requirements than other systems, mainly due to extensive use of free cooling. Cooling energy requirements of these two systems were much lower than associated building cooling demands in most of the simulated scenarios except for buildings with very low cooling requirements. The CAV system performed slightly better than the VAV system. The fan coil system and two chilled ceiling systems had much higher cooling energy requirements when compared to all air systems primarily due to two reasons: operating with constant fresh air supply which diminishes free cooling capabilities and additional heat gains from auxiliary equipment which need to be removed. These three systems performed very similar to each other with a major characteristic that their cooling requirements were always higher than associated building cooling demands.

Opposite to the cooling requirements, fan coil and chilled ceiling systems were systems with the lowest heating energy requirements which were caused by use of a heat recovery unit with relatively high efficiency of 65%. This affects ventilation losses which were significantly reduced. Other two systems had higher heating energy requirements although they were lower than associated building heating demands except in the low number of buildings equipped with the VAV system. The CAV system performed slightly better than the VAV system mainly due to two reasons. First is the type of the supply air temperature control. By varying the supply air temperature in the CAV system instead of keeping it constant as in the VAV system, the zone reheating requirements are reduced which is particularly important in situations when some zones require cooling while others request heating. The other reason is the higher level of additional heat gains from auxiliary equipment which reduces heat requirements.

Having a large dataset of different office buildings, for which the energy requirements of various HVAC systems were calculated, created an opportunity to investigate the possibility of developing a single coefficient which should represent a secondary HVAC system seasonal efficiency. The existence of such parameter would simplify the selection and comparison of HVAC systems. Following the analogy with primary systems, where such coefficients exist and they are widely used (boiler efficiency or chiller coefficient of performance), secondary HVAC system seasonal efficiency was calculated as a ratio between the annual building demand and the HVAC system energy requirement. Results of statistical analysis suggested that, unfortunately, there is no single parameter which can describe the secondary HVAC system seasonal efficiency. For all five investigated HVAC systems the values for both heating and cooling seasonal efficiencies were spread over a certain range. Fan coil system was closest to the single value for average seasonal cooling efficiency of 0.69 with the standard deviation of 0.028. All other systems had much higher standard deviations.

Another conclusion from the first part of the analysis was a strong correlation between HVAC systems heating/cooling energy requirements and office building heating/cooling demands. This became obvious when system heating/cooling energy requirements were plotted against building heating/cooling demand. These scatter plots also revealed that fan coil and chilled ceiling systems heating energy requirements are highly affected by the building floor space arrangement. While both HVAC systems heating and cooling energy requirements expressed strong correlation to building heating and cooling demands respectively, the HVAC systems auxiliary energy requirements did not correlate well with building total (heating + cooling) demands.

Analysis of the auxiliary energy requirements confirmed that the type of HVAC system drives the level of energy consumption. The CAV system had the highest auxiliary energy requirements, almost twice as high as the second largest consumer which was the VAV system. The fan coil system was slightly more economical than the VAV system, while chilled ceiling systems had the lowest consumption. The importance of including auxiliary energy consumptions in building energy end use analysis had been demonstrated by analysing the share of auxiliary energy requirements in buildings total demand. Average CAV system auxiliary energy requirements share in buildings total demand was close to 40% with a standard deviation of 10%, which is quite high. VAV and fan coil systems had average shares of 20% ± 5% and 17% ± 3% respectively, while chilled ceiling systems had an average share of 12% ± 3%. Additional analysis showed that on average, fans consume 93%, 80% and 75% of total auxiliary requirements in CAV, VAV and fan coil systems respectively with a relatively small standard deviation of up to 1.8%. On the other hand, fan and pump energy consumptions were nearly equal in chilled ceiling systems; 53% to 47% in advantage of fans with a standard deviation of 8.4%.

The second part of the analysis in this research dealt with the development of the regression models. Heating, cooling and auxiliary energy consumptions calculated by EnergyPlus for five different HVAC systems coupled with 3840 office buildings as well as building heating and cooling demands were fitted to two groups of statistical models. The first group included simple models based on the single independent variable, which was either building cooling demand or building heating demand. The second group was composed of models with two independent variables: heating and cooling demands. Outputs from the regression analysis were evaluated by inspecting models best fit parameter values and goodness of fit, which resulted in selecting the most appropriate model from each group of models.

The recommended simple regression model is based on the power function and it can be used to predict HVAC systems heating/cooling energy requirements as a function of building cooling/heating demands. The limitation of this model is that it cannot calculate HVAC systems auxiliary requirements. This model showed extraordinary good prediction of both heating and cooling energy requirements for all analysed systems. The CAV system had the lowest coefficients of determination which were 0.941 and 0.976 for cooling and heating energy requirement models respectively. All other systems had higher coefficients of determination including the fan coil system for which the power model provided the best fit with the R^2 above 0.99 for both cooling/heating models.

The complex model which provided the best fit was based on the full second order degree two polynomial function. This model provided even better overall fit than the simple model. The lowest R^2 was close to 0.98 in the case of the cooling energy requirements model and close to 0.99 in the case of the heating energy requirements model for all analysed HVAC systems, which can be assumed almost perfect fit. Auxiliary energy requirements of VAV, CAV and fan coil systems were also predicted very well with the R^2 between 0.95 and 0.98. On the other side, chilled ceiling systems auxiliary energy requirements did not fit as good as other systems, having the R^2 of 0.87, which can also be accepted as relatively high coefficient of determination.

Even though high coefficients of determination, which characterise almost all models, are an excellent measure of models overall goodness of fit, the analysis was further extended by evaluating the relative differences between the models‟ predicted and observed values. Analysis on relative differences provided additional information about both simple and complex model performances, although this approach has one major limitation; the relative error at low energy requirements can be enormous. Nevertheless, even taking into account this limitation, both recommended regression models showed acceptable behaviour. The simple model predicted cooling and heating energy requirements of all HVAC systems within ±20% relative difference range for more than 95% of all office buildings, except for the CAV system heating energy requirements for which almost 14% of data were outside the ±20% relative difference range. Between 55% and 85% of all data fitted within ±10% relative difference range, depending on the system, except for the fan coil cooling energy requirements model predictions which was almost 100% within ±10% relative difference range. Fan coil system simple cooling energy requirements model also fitted more than 90% of all data within ±5% relative difference range which is tremendous, especially if it is taken into account that all other system had only between 30% and 55% of both heating and cooling energy requirements predictions within this range.

The complex model showed even greater accuracy predicting the heating, cooling and auxiliary energy requirements with: above 99% within ±20% relative difference range for almost all HVAC systems, above 80% within ±10% relative difference range for almost all HVAC systems and above 50% within ±5% relative difference range for all systems.

Both high coefficients of determination, as a measure of models overall goodness of fit and quite acceptable relative differences proved that office building HVAC systems heating, cooling and auxiliary energy requirements can be predicted with a high accuracy by simplified regression models which are function of building heating and cooling demands.

In summary, the main conclusions are:

- All air systems (VAV and CAV) in general have lower cooling energy requirements than associated building cooling demands. The CAV system performs slightly better than the VAV system when responding to the office building cooling demand.
- All analysed HVAC systems have lower heating energy requirements than associated building heating demands. Fan coil system and chilled ceiling systems performs better than all air systems.
- The CAV systems have highest auxiliary energy requirements, followed by VAV and fan coil systems. Chilled ceiling systems are most energy efficient in terms of auxiliary energy.

- There is no single parameter which can present seasonal heating and cooling efficiency of the secondary HVAC systems as well as the share of auxiliary energy in total building demands, although certain efficiency ranges and associated standard deviations for each of the investigated systems were established.
- HVAC systems cooling and heating energy requirements correlate very well with office building cooling and heating demands respectively, while systems auxiliary requirements do not correlate on building total demands.
- Two simplified regression models which predict HVAC systems energy requirements as a function of building heating and cooling demands have been developed. The first model is based on the single independent variable (either building heating demand or building cooling demand) and can predict HVAC systems heating and cooling energy requirements with very high accuracy (the lowest R^2 was 0.94 for cooling model and 0.976 for heating model) but not auxiliary energy consumption. The second model is based on second order two independent variables polynomial function. This model can predict both heating and cooling energy requirements as well as auxiliary energy requirements as a function of building heating and cooling demands and has even higher accuracy than simple model. The lowest R^2 was 0.979 for cooling model and 0.983 for heating model, while the R^2 for auxiliary model was above 0.95 except in the case of chilled ceiling systems where the R^2 was around 0.87.

5.1. Further work

In addition to the significant findings this research delivered, it also raised several questions which can be further explored. Constraints of this research were presented in the last section of Chapter 4. Three categories of constraints were identified: office building dataset constraints, HVAC systems constraints and climate constraints.

Although the office building dataset, developed for the purpose of this research, is comprehensive, it can also be further extended by including buildings of more complex built form, light thermal mass, various air infiltration rates and different levels of internal gains. It would be also beneficial to place building in the urban area context which would evaluate the impact of shading from surrounding buildings.

Models of five HVAC systems analysed in this research were built according to standardised assumptions about their characteristics and operational control. Varying both system features and control features can have an impact on system thermal behaviour which potentially can affect the accuracy of developed regression models. In addition, HVAC systems can benefit from some advanced features such as night time ventilation, displacement ventilation or coupling with natural ventilation. These are only a few ideas how the impact of HVAC system on office buildings energy requirements can be further explored.

One of the major limitations of this research is the use of only one location weather data (London Gatwick) which means that the presented regression models are only suitable for buildings located in the southeast UK. Models cannot predict energy consumption of buildings located somewhere else which is an opportunity for further research. Subset of office buildings database was simulated with additional two weather files (Aberdeen and Finningley) and preliminary results suggested that it might be possible to develop one simple HVAC system heating energy consumption model as a function of building heating demand which would cover the whole UK. Unfortunately, preliminary results also revealed that it is not possible to develop an equivalent model for calculating cooling energy requirements. Most likely, introducing additional parameters in the cooling model would be required, such as cooling degree days or similar.

The conclusions about HVAC systems cooling and heating performance in this research were based on the analysis of secondary HVAC systems requirements with the assumption that there is an infinite amount of energy from ideal primary source (sources with efficiency 1) available all the time. The impact of the primary source was not taken into account. The step further would be coupling secondary HVAC systems with different types of primary sources in order to investigate the building overall environmental impact which is largely dependent on primary fuels types.

References

AHRI. 2011. *Directory of Certified Product Performance* [Online]. Available: http://www.ahridirectory.org.

AKSOY, U. T. & INALLI, M. 2006. Impacts of some building passive design parameters on heating demand for a cold region. *Building and Environment,* 41**,** 1742-1754.

AKTACIR, M. A., BÜYÜKALACA, O. & YILMAZ, T. 2006. Life-cycle cost analysis for constant-air-volume and variable-air-volume air-conditioning systems. *Applied Energy,* 83**,** 606-627.

ANSI/ASHRAE. 2004. Standard 55-2004 Thermal Environmental Conditions for Human Occupancy. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

ANSI/ASHRAE. 2007a. Standard 62.1-2007 Ventilation for Acceptable Indoor Air Quality. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

ANSI/ASHRAE. 2007b. Standard 90.1-2007 Energy Standard for Buildings Except Low-Rise Residential Buildings. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

ANSI/ASHRAE. 2007c. Standard 140-2007 Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

ARDEHALI, M. M. & SMITH, T. F. 1996. Evaluation of variable volume and temperature HVAC system for commercial and residential buildings. *Energy Conversion and Management,* 37**,** 1469-1479.

ASAN, H. 1998. Effects of wall's insulation thickness and position on time lag and decrement factor. *Energy and Buildings,* 28**,** 299-305.

ASAN, H. 2000. Investigation of wall's optimum insulation position from maximum time lag and minimum decrement factor point of view. *Energy and Buildings,* 32**,** 197-203.

ASHRAE 2007. *Handbook 2007: HVAC Applications*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

ASHRAE 2008. *Handbook 2008: HVAC Systems and Equipment*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

ASHRAE 2009. *Handbook 2009: Fundamentals*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

ASTE, N., ANGELOTTI, A. & BUZZETTI, M. 2009. The influence of the external walls thermal inertia on the energy performance of well insulated buildings. *Energy and Buildings,* 41**,** 1181-1187.

AYNUR, T. N., HWANG, Y. & RADERMACHER, R. 2009. Simulation comparison of VAV and VRF air conditioning systems in an existing building for the cooling season. *Energy and Buildings,* 41**,** 1143-1150.

BALARAS, C. A. 1996. The role of thermal mass on the cooling load of buildings. An overview of computational methods. *Energy and Buildings,* 24**,** 1-10.

BANSAL, N. K. & BHATTACHARYA, A. 2009. Parametric equations for energy and load estimations for buildings in India. *Applied Thermal Engineering,* 29**,** 3710-3715.

BCO. 2009a. Occupier Density Study - Summary Report. British Council for Offices, London, UK.

BCO. 2009b. Small Power Use in Offices. British Council for Offices, London, UK.

BODART, M. & DE HERDE, A. 2002. Global energy savings in offices buildings by the use of daylighting. *Energy and Buildings,* 34**,** 421-429.

BOKEL, R. M. J. 2007. The effect of window position and window size on the energy demand for heating, cooling and electric lighting. *10th Int. IBPSA Conference - BS 2007.* Beijing (China).

BOLATTÜRK, A. 2006. Determination of optimum insulation thickness for building walls with respect to various fuels and climate zones in Turkey. *Applied Thermal Engineering,* 26**,** 1301-1309.

BOLATTÜRK, A. 2008. Optimum insulation thicknesses for building walls with respect to cooling and heating degree-hours in the warmest zone of Turkey. *Building and Environment,* 43**,** 1055-1064.

BRAHAM, D., BARNARD, N. & JAUNZENS, D. 2001. Thermal mass in office buildings BRE Digest 454 Parts 1 and 2 (Garston: Building Research Establishment).

BRANDEMUEHL, M. J. 1993. *HVAC 2 Toolkit: AToolkit for Secondary HVAC System Energy Calculations*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

BRAUN, J. E. 2003. Load control using building thermal mass. *Journal of Solar Energy Engineering, Transactions of the ASME,* 125**,** 292-301.

BRAUN, J. E. & ZHONG, Z. 2005. Development and evaluation of a night ventilation precooling algorithm. *HVAC and R Research,* 11**,** 433-458.

BRE. 2004. Good Building Gude 61 - Lighting: non domestic. Building Research Establishment, UK.

BRESCU. 1996. General Information Report GIR 40: Heating systems and their control. Building Research Energy Conservation Support Unit, Garston, UK.

BROWN, F. E., RICKABY, P. A., BRUHNS, H. R. & STEADMAN, P. 2000. Surveys of nondomestic buildings in four English towns. *Environment and Planning B: Planning and Design,* 27**,** 11-24.

BSRIA. 2008. *Air conditioning - UK market review* [Online]. Available: http://www.bsria.co.uk/news/1684/.

BSRIA. 2009. *UK air conditioning market overview* [Online]. Available: http://www.bsria.co.uk/news/ac-market/.

CAEIRO, J., ALTAMIRANO-MEDINA, H., BRUHNS, H. & ALTAN, H. 2008. Using street surveys to establishg air conditioning incidence in UK's commercial offices stock. *International Conference on Improving Energy Efficiency in Commercial Buildings "IEECB'08".* Frankfurt (Germany).

CARROLL, W. L. & HITCHCOCK, R. J. 2005. DElight2 Daylighting Analysis in Energy Plus: Integration and Preliminary User Results. *9th Int. IBPSA Conference - BS 2005.* Montreal (Canada).

CEN. 2002. EN 12464-1:2002 Light and lighting - Lighting of work places - Part 1: Indoor work places. Brussels (Belgium).

CEN. 2007a. EN 13779:2007 Ventilation for non-residential buildings - Performance requirements for ventilation and room-conditioning systems. Brussels (Belgium).

CEN. 2007b. EN 15251:2007 Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics. Brussels (Belgium).

CHANTRASRISALAI, C., GHATTI, V., FISHER, D. E. & SCHEATZLE, D. G. 2003. Experimental validation of the EnergyPlus low-temperature radiant simulation. *ASHRAE Transactions Technical and Symposium Papers.* Kansas City, MO.

CHO, Y. H. & LIU, M. 2009a. Minimum airflow reset of single duct VAV terminal boxes. *Building and Environment,* 44**,** 1876-1885.

CHO, Y. H. & LIU, M. 2009b. Single-duct constant volume system optimization. *Energy and Buildings,* 41**,** 856-862.

CHO, Y. H., LIU, M., WANG, G., WANG, J. & RAUSCHER, T. 2009. VFD application for constant volume air-handling units. *ASHRAE Transactions.* Louisville, KY.

CHVATAL, K. M. S. & CORVACHO, H. 2009. The impact of increasing the building envelope insulation upon the risk of overheating in summer and an increased energy consumption. *Journal of Building Performance Simulation,* 2**,** 267-282.

CIBSE 2004. *Guide F: Energy efficiency in buildings*, Chartered Institution of Building Services Engineers, London, UK.

CIBSE 2005a. *Guide B: Heating, Ventilating, Air Conditioning and Refrigeration*, Chartered Institution of Building Services Engineers, London, UK.

CIBSE. 2005b. Lighting Guide 7: Office Lighting. Chartered Institution of Building Services Engineers, London, UK.

CIBSE 2006. *Guide A: Environmental design*, Chartered Institution of Building Services Engineers, London, UK.

CIBSE 2009. *Guide H: Building Control Systems*, Chartered Institution of Building Services Engineers, London, UK.

CORDOBA, J., MACIAS, M. & ESPINOSA, J. M. 1998. Study of the potential savings on energy demand and HVAC energy consumption by using coated glazing for office buildings in Madrid. *Energy and Buildings,* 27**,** 13-19.

CRAWLEY, D. B. 1998. Which weather data should you use for energy simulations of commercial buildings? *Proceedings of the 1998 ASHRAE Annual Meeting.* 2 ed. Toronto, Canada.

CRAWLEY, D. B., HAND, J. W., KUMMERT, M. & GRIFFITH, B. T. 2008. Contrasting the capabilities of building energy performance simulation programs. *Building and Environment,* 43**,** 661-673.

CRAWLEY, D. B., LAWRIE, L. K., WINKELMANN, F. C., BUHL, W. F., HUANG, Y. J., PEDERSEN, C. O., STRAND, R. K., LIESEN, R. J., FISHER, D. E., WITTE, M. J. & GLAZER, J. 2001a. EnergyPlus: Creating a new-generation building energy simulation program. *Energy and Buildings,* 33**,** 319-331.

CRAWLEY, D. B., LAWRIE, L. K., WINKELMANN, F. C. & PEDERSEN, C. O. 2001b. Energyplus: new capabilities in a whole-building energy simulation program. *7th Int. IBPSA Conference - BS 2001.* Rio de Janeiro (Brazil).

DASCALAKI, E. & SANTAMOURIS, M. 2002. On the potential of retrofitting scenarios for offices. *Building and Environment,* 37**,** 557-567.

DCLG. 2007. Report on carbon reduction in new non-domestic buildings. Department for Communities and Local Government, London, UK.

DCLG. 2009. Floorspace and rateable value of commercial and industrial properties 1 April 2008, (England & Wales). Department for Communities and Local Government, London, UK.

DCLG. 2010. Non-domestic Building Services Compliance Guide - 2010 Edition. Department for Communities and Local Government, London, UK.

DCLG. 2011. A Technical Manual for SBEM. Department for Communities and Local Government, London, UK.

DESIGNBUILDER. v.2. Material Database. http://www.designbuilder.co.uk/.

DEWO. 1985. The Building Regulations 1985 - Approved Document L: Conservation of fuel and power (1985 edition). Department of the Environment and the Welsh Office, London, UK.

DEWO. 1990. The Building Regulations 1985 - Approved Document L: Conservation of fuel and power (1990 edition). Department of the Environment and the Welsh Office, London, UK.

DEWO. 1995. The Building Regulations 1991 - Approved Document L: Conservation of fuel and power (1995 edition). Department of the Environment and the Welsh Office, London, UK.

DOMBAYCI, Ö. A. 2007. The environmental impact of optimum insulation thickness for external walls of buildings. *Building and Environment,* 42**,** 3855-3859.

DOMBAYCI, Ö. A., GÖLCÜ, M. & PANCAR, Y. 2006. Optimization of insulation thickness for external walls using different energy-sources. *Applied Energy,* 83**,** 921-928.

DTLR. 2002a. The Building Regulations 2000 - Approved Document L1: Conservation of fuel and power - dwellings (2002 edition). Department for Transport, Local Government and the Regions, London, UK.

DTLR. 2002b. The Building Regulations 2000 - Approved Document L2: Conservation of fuel and power in buildings other than dwellings (2002 edition). Department for Transport, Local Government and the Regions, London, UK.

DUNN, G. & KNIGHT, I. 2005. Small power equipment loads in UK office environments. *Energy and Buildings,* 37**,** 87-91.

EASTOP, T. D. & WATSON, W. E. 1992. *Mechanical Services for Buildings*, Longman Group (FE) Ltd.

EC. 2003. DIRECTIVE 2002/91/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 16 December 2002 on the energy performance of buildings. Brussels, 16 December 2002.

ECG019. 2003. Energy Efficiency-Best Practice Programme – Energy Consumption Guide 019: Energy Use in Offices. Carbon Trust.

ELLIS, M. W. & MATHEWS, E. H. 2002. Needs and trends in building and HVAC system design tools. *Building and Environment,* 37**,** 461-470.

EMMERICH, S. J., MCDOWELL, T. P. & ANIS, W. 2007. Simulation of the impact of commercial building envelope airtightness on building energy utilization. *ASHRAE Transactions.* Long Beach, CA.

EMMERICH, S. J. & PERSILY, A. K. 1998. Energy Impacts of Infiltration and Ventilation in U.S. Office Buildings Using Multizone Airflow Simulation. *IAQ and Energy 98.* New Orleans, LA.

EMMERICH, S. J. & PERSILY, A. K. 2005. Airtightness of commercial buildings in the U.S. *26th AIVC Conference.* Brussels (Belgium).

ENERGYPLUS. 2010a. *Auxiliary EnergyPlus Programs* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/pdfs/auxiliaryprograms.pdf.

ENERGYPLUS. 2010b. *Energy Simulation Software v.6.0.0* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/.

ENERGYPLUS. 2010c. *EnergyPlus Testing with ASHRAE 1052-RP Toolkit – Building Fabric Analytical Tests* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/pdfs/energyplus_ashrae_1052rp_envelope.pdf.

ENERGYPLUS. 2010d. *EnergyPlus Testing with Building Thermal Envelope and Fabric Load Tests from ANSI/ASHRAE Standard 140-2007* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/pdfs/energyplus_ashrae_140_envelope.pdf.

ENERGYPLUS. 2010e. *EnergyPlus Testing with HVAC Equipment Performance Tests CE100 to CE200 from ANSI/ASHRAE Standard 140-2007* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/pdfs/energyplus_ashrae_140_hvac_ce100to200.pdf.

ENERGYPLUS. 2010f. *EnergyPlus Testing with HVAC Equipment Performance Tests CE300 to CE545 from ANSI/ASHRAE Standard 140-2007* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/pdfs/energyplus_ashrae_140_hvac_ce300to545.pdf.

ENERGYPLUS. 2010g. *EnergyPlus Testing with IEA BESTEST Mechanical Equipment & Control Strategies for a Chilled Water and a Hot Water System* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/pdfs/energyplus_hvac_heat-cool-coil_tests.pdf.

ENERGYPLUS. 2010h. *EnergyPlus Testing with IEA BESTEST Multi-Zone NonAirflow In-Depth Diagnostic Cases MZ320 – MZ360* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/pdfs/energyplus_multiizone-non-airflow_tests.pdf.

ENERGYPLUS. 2010i. *Getting Started with EnergyPlus - Basic Concepts Manual* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/pdfs/gettingstarted.pdf.

ENERGYPLUS. 2010j. *Weather Data* [Online]. U.S. Department of Energy. Available: http://apps1.eere.energy.gov/buildings/energyplus/weatherdata_about.cfm.

ENGDAHL, F. & JOHANSSON, D. 2004. Optimal supply air temperature with respect to energy use in a variable air volume system. *Energy and Buildings,* 36**,** 205-218.

FELSMANN, C. 2008. Interantion Energy Agency Task 34: Mechanical Equipment & Control Strategies for a Chilled water and a Hot water system Technical University of Dresden, Germany.

GAKOVIC, B. 2000. Areas and types of glazing and other openings in the nondomestic building stock. *Environment and Planning B: Planning and Design,* 27**,** 667-694.

GOV.LEGISLATION. 1965. The Building Regulations 1965 - Statutory Instruments, vol. 1373.

GOV.LEGISLATION. 1972. The Building Regulations 1972 - Statutory Instruments, vol. 317.

GOV.LEGISLATION. 1976. The Building Regulations 1976 - Statutory Instruments, vol. 1676.

HUI, S. C. M. 1997. A randomised approach to multiple regression analysis of building energy simulation. *5th Int. IBPSA Conference - BS 1997.* Prague (Czech Republic).

IEA. 2007. IEA SHC Task 34 - Testing and Validation of Building Energy Simulation Tools.

IES 1993. *Lighting Handbook: Reference & Application*, Illuminating Engineering Society of North America, New York, USA.

JAFFAL, I., INARD, C. & GHIAUS, C. 2009. Fast method to predict building heating demand based on the design of experiments. *Energy and Buildings,* 41**,** 669-677.

JENKINS, D., LIU, Y. & PEACOCK, A. D. 2008. Climatic and internal factors affecting future UK office heating and cooling energy consumptions. *Energy and Buildings,* 40**,** 874-881.

JENKINS, D. P. 2009. The importance of office internal heat gains in reducing cooling loads in a changing climate. *International Journal of Low-Carbon Technologies,* 4**,** 134-140.

JENKINS, D. P., SINGH, H. & EAMES, P. C. 2009. Interventions for large-scale carbon emission reductions in future UK offices. *Energy and Buildings,* 41**,** 1374-1380.

JUDKOFF, R. & NEYMARK, J. 1995. Interantion Energy Agency Task 12: Building Energy Simulation Test (BESTEST) and Diagnostic Methods. National Renewable Energy Laboratory, Golden, CO.

KALAISELVAM, S., VELICHET, V. S., INIYAN, S. & SAMUEL, A. A. 2006. Comparative energy analysis of a constant air volume (CAV) system and a variable air volume (VAV) system for a software laboratory. *International Journal of Ventilation,* 5**,** 229-237.

KE, Y. P. & MUMMA, S. A. 1997. Optimized supply-air temperature (SAT) in variable-air-volume (VAV) systems. *Energy,* 22**,** 601-614.

KIM, J.-J. & MOON, J. W. 2009. Impact of insulation on building energy consumption. *11th Int. IBPSA Conference - BS 2009.* Glasgow (UK).

KNIGHT, I. & DUNN, G. 2005. Measured energy consumption and carbon emissions of air-conditioning in UK office buildings. *Building Services Engineering Research and Technology,* 26**,** 89-98.

KNIGHT, I. P. 1999. Measured energy savings due to photocell control of individual luminaires. *Lighting Research and Technology,* 31**,** 19-22.

KOLOKOTRONI, M. & ARONIS, A. 1999. Cooling-energy reduction in air-conditioned offices by using night ventilation. *Applied Energy,* 63**,** 241-253.

KOLOKOTRONI, M., WEBB, B. C. & HAYES, S. D. 1998. Summer cooling with night ventilation for office buildings in moderate climates. *Energy and Buildings,* 27**,** 231-237.

KOROLIJA, I., MARJANOVIC-HALBURD, L., ZHANG, Y. & HANBY, V. I. 2011. Influence of building parameters and HVAC systems coupling on building energy performance. *Energy and Buildings,* 43**,** 1247-1253.

KOROLIJA, I., ZHANG, Y., MARJANOVIC-HALBURD, L. & HANBY, V. I. 2009. Selecting HVAC systems for typical UK office buildings. *6th International Symposium on Heating, Ventilating and Air Conditioning, ISHVAC 2009.* Nanjing (China).

KOSSECKA, E. & KOSNY, J. 2002. Influence of insulation configuration on heating and cooling loads in a continuously used building. *Energy and Buildings,* 34**,** 321-331.

KREIDER, J. F. 2001. *Handbook of heating, ventilation, and air conditioning,* Boca Raton; London, CRC Press.

KUSIAK, A., LI, M. & TANG, F. 2010. Modeling and optimization of HVAC energy consumption. *Applied Energy,* 87**,** 3092-3102.

LAM, J. C., HUI, S. C. M. & CHAN, A. L. S. 1997. Regression analysis of high-rise fully air-conditioned office buildings. *Energy and Buildings,* 26**,** 189-197.

LAM, J. C. & LI, D. H. W. 1998. Daylighting and energy analysis for air-conditioned office buildings. *Energy,* 23**,** 79-89.

LAM, J. C., WAN, K. K. W., LIU, D. & TSANG, C. L. 2010. Multiple regression models for energy use in air-conditioned office buildings in different climates. *Energy Conversion and Management,* 51**,** 2692-2697.

LEE, E. S. & SELKOWITZ, S. E. 2006. The New York Times Headquarters daylighting mockup: Monitored performance of the daylighting control system. *Energy and Buildings,* 38**,** 914-929.

LI, D. H. W. & LAM, J. C. 2001. Evaluation of lighting performance in office buildings with daylighting controls. *Energy and Buildings,* 33**,** 793-803.

LI, D. H. W., LAM, T. N. T. & WONG, S. L. 2006. Lighting and energy performance for an office using high frequency dimming controls. *Energy Conversion and Management,* 47**,** 1133-1145.

LOUTZENHISER, P. G., MANZ, H., FELSMANN, C., STRACHAN, P. A., FRANK, T. & MAXWELL, G. M. 2007a. Empirical validation of models to compute solar irradiance on inclined surfaces for building energy simulation. *Solar Energy,* 81**,** 254-267.

LOUTZENHISER, P. G., MANZ, H., FELSMANN, C., STRACHAN, P. A. & MAXWELL, G. M. 2007b. An empirical validation of modeling solar gain through a glazing unit with external and internal shading screens. *Applied Thermal Engineering,* 27**,** 528-538.

LOUTZENHISER, P. G., MANZ, H., MOOSBERGER, S. & MAXWELL, G. M. 2009. An empirical validation of window solar gain models and the associated interactions. *International Journal of Thermal Sciences,* 48**,** 85-95.

LOUTZENHISER, P. G., MANZ, H., STRACHAN, P. A., FELSMANN, C., FRANK, T., MAXWELL, G. M. & OELHAFEN, P. 2006. An empirical validation of modeling solar gains through a glazing unit using building energy simulation programs. *HVAC and R Research,* 12**,** 1097-1116.

LU, L., CAI, W., CHAI, Y. S. & XIE, L. 2005a. Global optimization for overall HVAC systems - Part I problem formulation and analysis. *Energy Conversion and Management,* 46**,** 999-1014.

LU, L., CAI, W., SOH, Y. C. & XIE, L. 2005b. Global optimization for overall HVAC systems - Part II problem solution and simulations. *Energy Conversion and Management,* 46**,** 1015-1028.

LU, L., CAI, W., XIE, L., LI, S. & SOH, Y. C. 2005c. HVAC system optimization - In-building section. *Energy and Buildings,* 37**,** 11-22.

MAAMARI, F., ANDERSEN, M., DE BOER, J., CARROLL, W. L., DUMORTIER, D. & GREENUP, P. 2006. Experimental validation of simulation methods for bi-directional transmission properties at the daylighting performance level. *Energy and Buildings,* 38**,** 878-889.

MANZAN, M. & PINTO, F. 2009. Genetic optimization of external shading devices. *11th Int. IBPSA Conference - BS 2009.* Glasgow (UK).

MARJANOVIC-HALBURD, L., KOROLIJA, I. & HANBY, V. 2008. Heating Ventilating and Air-Conditioning (HVAC) Equipment Taxonomy. *IIR 2008 - HVAC Energy Efficiency Best Practice Conference.* Melbourne (Australia).

MASOSO, O. T. & GROBLER, L. J. 2008. A new and innovative look at anti-insulation behaviour in building energy consumption. *Energy and Buildings,* 40**,** 1889-1894.

MORTIMER, N. D., ASHLEY, A. & RIX, J. H. R. 2000. Detailed energy surveys of nondomestic buildings. *Environment and Planning B: Planning and Design,* 27**,** 25-32.

MUMMA, S. A. 2002. Chilled ceilings in parallel with dedicated outdoor air systems: Addressing the concerns of condensation, capacity, and cost. *ASHRAE Transactions 2002.* Honolulu, HI.

NETO, A. H. & FIORELLI, F. A. S. 2008. Comparison between detailed model simulation and artificial neural network for forecasting building energy consumption. *Energy and Buildings,* 40**,** 2169-2176.

NEYMARK, J. & JUDKOFF, R. 2002. Interantion Energy Agency Task 22: International Energy Agency Building Energy Simulation Test and Diagnostic Method for Heating, Ventilating, and Air-Conditioning Equipment Models (HVAC BESTEST) Volume 1: Cases E100–E200. National Renewable Energy Laboratory, Golden, CO.

NEYMARK, J. & JUDKOFF, R. 2004. Interantion Energy Agency Task 22: International Energy Agency Building Energy Simulation Test and Diagnostic Method for Heating, Ventilating, and Air-Conditioning Equipment Models (HVAC BESTEST) Volume 2: Cases E300–E545. National Renewable Energy Laboratory, Golden, CO.

NEYMARK, J. & JUDKOFF, R. 2008. Interantion Energy Agency Task 34: International Energy Agency Building Energy Simulation Test and Diagnostic Method (IEA BESTEST): Multi-Zone Non-Airflow In-Depth Diagnostic Cases: MZ320 – MZ360. National Renewable Energy Laboratory, Golden, CO.

NIU, J., KOOI, J. V. D. & RHEE, H. V. D. 1995. Energy saving possibilities with cooled-ceiling systems. *Energy and Buildings,* 23**,** 147-158.

NOVOSELAC, A. & SREBRIC, J. 2002. A critical review on the performance and design of combined cooled ceiling and displacement ventilation systems. *Energy and Buildings,* 34**,** 497-509.

ODPM. 2005. Age of Commercial and Industrial Stock: Local Authority Level 2004 (England and Wales). Office of the Deputy Prime Minister, London, UK.

ODPM. 2006a. The Building Regulations 2000 - Approved Document F: Ventilation (2006 edition). Office of the Deputy Prime Minister, London, UK.

ODPM. 2006b. The Building Regulations 2000 - Approved Document L2A: Conservation of fuel and power (New buildings other than dwellings) (2006 edition). Office of the Deputy Prime Minister, London, UK.

ODPM. 2006c. Commercial and Industrial Floorspace and Rateable Value Statistics 2005 (2005 Revaluation). Office of the Deputy Prime Minister, London, UK.

ODPM. 2010. The Building Regulations 2000 - Approved Document L2A: Conservation of fuel and power in new buildings other than dwellings (2010 edition). Office of the Deputy Prime Minister, London, UK.

OUGHTON, D. R. & HODKINSON, S. L. 2008. *Faber & Kell's Heating and Air-Conditioning of Buildings*, Faber Maunsell. Published by Elsevier Ltd.

OURGHI, R., AL-ANZI, A. & KRARTI, M. 2007. A simplified analysis method to predict the impact of shape on annual energy use for office buildings. *Energy Conversion and Management,* 48**,** 300-305.

OZEL, M. & PIHTILI, K. 2007. Optimum location and distribution of insulation layers on building walls with various orientations. *Building and Environment,* 42**,** 3051-3059.

PÉREZ-LOMBARD, L., ORTIZ, J. & POUT, C. 2008. A review on buildings energy consumption information. *Energy and Buildings,* 40**,** 394-398.

PERSILY, A. K. 1998. Airtightness of Commercial and Institutional Buildings: Blowing Holes in the Myth of Tight Buildings. *Thermal Envelopes VII Conference.* Clearwater, FL.

PITA, E. G. 2002. *Air conditioning principles and systems,* Upper Saddle River, N.J.; London, Prentice Hall.

POIRAZIS, H., BLOMSTERBERG, A. & WALL, M. 2008. Energy simulations for glazed office buildings in Sweden. *Energy and Buildings,* 40**,** 1161-1170.

RAEISSI, S. & TAHERI, M. 1998. Optimum overhang dimensions for energy saving. *Building and Environment,* 33**,** 293-302.

RENNIE, D. & PARAND, F. 1998. Environmental design guide for naturally ventilated and daylit offices. BRE Report 345 (Garston: Construction Research Communications Ltd.).

RICKABY, P. A. & GORGOLEWSKI, M. T. 2000. A classification system for services in nondomestic buildings. *Environment and Planning B: Planning and Design,* 27**,** 695-708.

RICS. 2007. Code of Measuring Practice - A guide for Property Professionals (6th edition). Royal Institution of Chartered Surveyors, London, UK.

ROISIN, B., BODART, M., DENEYER, A. & D'HERDT, P. 2008. Lighting energy savings in offices using different control systems and their real consumption. *Energy and Buildings,* 40**,** 514-523.

SANDER, D. M., CORNICK, S., NEWSHAM, G. R. & CRAWLEY, D. B. 1993. Development of a Simple Model to Relate Heating and Cooling Energy to Building Envelope Thermal Characteristics. *3rd Int. IBPSA Conference - BS 1993.* Adelaide (Australia).

SCHEATZLE, D. G. 2003. Data set for validating simulation tools for radiant/convective systems. *ASHRAE Transactions Technical and Symposium Papers.* Kansas City, MO.

SEKHAR, S. C. 1997. A critical evaluation of variable air volume system in hot and humid climates. *Energy and Buildings,* 26**,** 223-232.

SEKHAR, S. C. & YAT, C. J. 1998. Energy simulation approach to air-conditioning system evaluation. *Building and Environment,* 33**,** 397-408.

SEPPÄNEN, O., FISK, W. J. & LEI, Q. H. 2006. Ventilation and performance in office work. *Indoor Air,* 16**,** 28-36.

SEPPÄNEN, O. A. & FISK, W. 2006. Some quantitative relations between indoor environmental quality and work performance or health. *HVAC and R Research,* 12**,** 957-973.

SEPPÄNEN, O. A., FISK, W. J. & MENDELL, M. J. 1999. Association of ventilation rates and CO2 concentrations with health and other responses in commercial and institutional buildings. *Indoor Air,* 9**,** 226-252.

SEZGEN, O. & KOOMEY, J. G. 2000. Interactions between lighting and space conditioning energy use in US commercial buildings. *Energy,* 25**,** 793-805.

SHARPLES, S., CLOSS, S. & CHILENGWE, N. 2005. Airtightness testing of very large buildings: A case study. *Building Services Engineering Research and Technology,* 26**,** 167-172.

SODEC, F. 1999. Economic viability of cooling ceiling systems. *Energy and Buildings,* 30**,** 195-201.

SPITLER, J. D., REES, S. J. & DONGYI, X. 2001. Development of an analytical verification test suite for whole building energy simulation programs - Building fabric. Oklahoma State University, School of Mechanical and Aerospace Engineering, OK.

STEADMAN, P., BRUHNS, H. R., HOLTIER, S., GAKOVIC, B., RICKABY, P. A. & BROWN, F. E. 2000a. A classification of built forms. *Environment and Planning B: Planning and Design,* 27**,** 73-91.

STEADMAN, P., BRUHNS, H. R. & RICKABY, P. A. 2000b. An introduction to the national Non-Domestic Building Stock database. *Environment and Planning B: Planning and Design,* 27**,** 3-10.

STEEMERS, K. 2003. Energy and the city: Density, buildings and transport. *Energy and Buildings,* 35**,** 3-14.

STEGOU-SAGIA, A., ANTONOPOULOS, K., ANGELOPOULOU, C. & KOTSIOVELOS, G. 2007. The impact of glazing on energy consumption and comfort. *Energy Conversion and Management,* 48**,** 2844-2852.

STOCKI, M., CURCIJA, D. C. & BHANDARI, M. S. 2007. The development of standardized whole-building simulation assumptions for energy analysis for a set of commercial buildings. *ASHRAE Transactions.* Long Beach, CA.

TIAN, Z. & LOVE, J. A. 2005. An integrated study of radiant slab cooling systems through experiment and building simulation. *9th Int. IBPSA Conference - BS 2005.* Montreal (Canada).

TIAN, Z. & LOVE, J. A. 2009. Energy performance optimization of radiant slab cooling using building simulation and field measurements. *Energy and Buildings,* 41**,** 320-330.

TZEMPELIKOS, A. & ATHIENITIS, A. K. 2005. Integrated thermal and daylighting analysis for design of office buildings. *ASHRAE Transactions.* Orlando, FL.

U.S.DOE. 2009. Buildings Energy Data Book. U.S. Department of Energy.

VANBRONKHORST, D. A., PERSILY, A. K. & EMMERICH, S. J. 1995. Energy Impacts of Air Leakage in U.S. Office Buildings. *16th AIVC Conference.* Palm Springs, CA.

WANG, S. & MA, Z. 2008. Supervisory and optimal control of building HVAC systems: A review. *HVAC and R Research,* 14**,** 3-32.

WANG, S. K. 2000. *Handbook of air conditioning and refrigeration,* New York; London, McGraw-Hill.

WARGOCKI, P., WYON, D. P., SUNDELL, J., CLAUSEN, G. & FANGER, P. O. 2000. The effects of outdoor air supply rate in an office on perceived air quality, sick building syndrome (SBS) symptoms and productivity. *Indoor Air,* 10**,** 222-236.

WESTPHALEN, D. & KOSZALINSKI, S. 1999. Energy Consumption Characteristics of Commercial Building HVAC Systems Volume II: Thermal Distribution, Auxiliary Equipment, and Ventilation. U.S. Department of Energy.

WRIGHT, J. & ZHANG, Y. 2008. The minimum capacity of HVAC secondary systems (with capacity reduction by interzonal airflow) (RP-1049). *HVAC and R Research,* 14**,** 397-415.

YANG, K. H. & TING, M. M. 1999. An innovative analysis and experimental investigation on energy savings of a VAV system in hot and humid climates. *Building and Environment,* 35**,** 27-31.

YAO, Y., LIAN, Z., LIU, W., HOU, Z. & WU, M. 2007. Evaluation program for the energy-saving of variable-air-volume systems. *Energy and Buildings,* 39**,** 558-568.

YAO, Y. & WANG, L. 2010. Energy analysis on VAV system with different air-side economizers in China. *Energy and Buildings,* 42**,** 1220-1230.

ZHANG, Y. 2009. "Parallel" EnergyPlus and the development of a parametric analysis tool. *11th Int. IBPSA Conference - BS 2009.* Glasgow (UK).

ZHANG, Y. & KOROLIJA, I. 2010. Performing complex parametric simulations with jEPlus. *SET 2010 - 9th International Conference on Sustainable Energy Technologies.* Shanghai (China).

ZHANG, Y., WRIGHT, J. A. & HANBY, V. I. 2006. Energy aspects of HVAC system configurations - Problem definition and test cases. *HVAC and R Research,* 12**,** 871-888.

ZHOU, Y. P., WU, J. Y., WANG, R. Z. & SHIOCHI, S. 2007. Energy simulation in the variable refrigerant flow air-conditioning system under cooling conditions. *Energy and Buildings,* 39**,** 212-220.

ZMEUREANU, R. & PERAGINE, C. 1999. Evaluation of interactions between lighting and HVAC systems in a large commercial building. *Energy Conversion and Management,* 40**,** 1229-1236.

Papers by the Author

Journal papers:

KOROLIJA, I., MARJANOVIC-HALBURD, L., ZHANG, Y. & HANBY, V. I. 2011. Influence of building parameters and HVAC systems coupling on building energy performance. *Energy and Buildings*, 43, 1247-1253.

Conference papers – refereed:

ZHANG, Y. & **KOROLIJA, I.** 2010. Performing complex parametric simulations with jEPlus. *SET 2010 - 9th International Conference on Sustainable Energy Technologies*. Shanghai (China).

LABUS, J., **KOROLIJA, I.**, MARJANOVIC-HALBURD, L. & CORONAS, A. 2010. The Influence of Weather Conditions on Energy Performance of HVAC System and Absorption Cooling System Coupling. *IAQVEC 2010 - The 7th International Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings*. Syracuse, NY.

KOROLIJA, I., HANBY, V. I., ZHANG, Y. & MARJANOVIC-HALBURD, L. 2010. HVAC Systems Energy Demand vs. Building Energy Demand. *1st Annual IESD PhD Conference: Energy and Sustainable Development*. Leicester (UK).

KOROLIJA, I., ZHANG, Y., MARJANOVIC-HALBURD, L. & HANBY, V. I. 2009. Selecting HVAC systems for typical UK office buildings. *6th International Symposium on Heating, Ventilating and Air Conditioning, ISHVAC 2009*. Nanjing (China).

KOROLIJA, I., HANBY, V., MARJANOVIC-HALBURD, L. & ZHANG, Y. 2008. A Comparison of the ASHRAE Secondary HVAC Toolkit Detailed and Simple Cooling Coil Models with Manufacturers' Data. *IIR 2008 - HVAC Energy Efficiency Best Practice Conference*. Melbourne (Australia).

MARJANOVIC-HALBURD, L., **KOROLIJA, I.** & HANBY, V. 2008. Heating Ventilating and Air-Conditioning (HVAC) Equipment Taxonomy. *IIR 2008 - HVAC Energy Efficiency Best Practice Conference*. Melbourne (Australia).

MARJANOVIC-HALBURD, L., **KOROLIJA, I.**, LIDDIARD, R. & WRIGHT, A. 2008. Operational Rating vs Asset Rating vs Detailed Simulation. *International Conference on Improving Energy Efficiency in Commercial Buildings "IEECB'08"*. Frankfurt (Germany).

Appendix A. Construction Elements

This appendix describes the physical properties and configuration for the building envelope and interior elements, which are walls, roofs, floors, windows and internal partitions.

Building element constructions in EnergyPlus are defined by the composition of material layers. Layers are specified by basic thermal and other material property parameters. The most important parameters are thickness, conductivity, density, and specific heat.

Building envelope elements in the models are:

- ground floor,
- flat roof, and
- external wall.

Figure A.1 shows the cross section of the ground floor element which is composed of five layers: clay underfloor (750mm), brick slips (25mm), cast concrete (100mm), insulation layer, and flooring screed (50mm). Insulation material used in ground floor is expanded polystyrene (EPS). The thickness of insulation layer was selected to follow the improvements in building regulations regarding the conversation

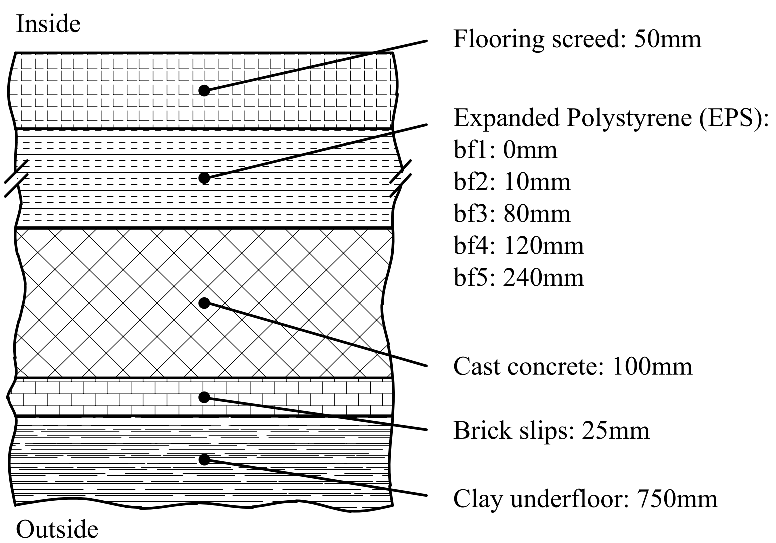


Figure A.1. Ground floor – cross section

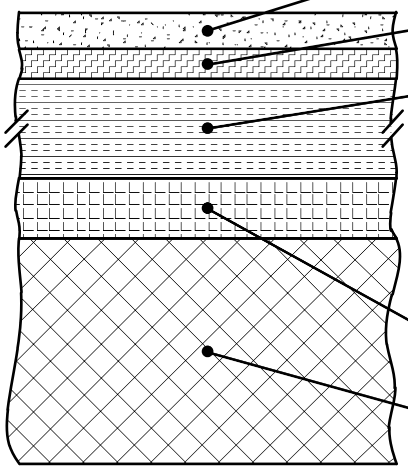


Figure A.2. Flat roof – cross section

of energy, starting with no insulation in the case of the building fabric 1 (BF1) construction type up to 240mm thick insulation for the best practice building fabric (BF5).

Flat roof construction element, Figure A.2, is built from following components: concrete-dense (150mm), flooring screed (40mm), insulation layer, asphalt (19mm), and stone chippings (25mm). Mineral wool – dense was used as insulation material, which thickness varied between 70mm for the building fabric type 2, and 240mm for the building fabric type 5. In the case of building fabric type 1 there was no insulation layer.

External wall, as it can be seen from Figure A.3, is also made from five layers: brick outer leaf (105mm), insulation layer, concrete 1800 (100mm), air cavity, and plasterboard (13mm) as interior surface finish. In the construction with no insulation (building fabric type 1), the insulation layer was replaced with air cavity. The thickness of insulation material (mineral wool quilt) in the rest of building fabrics increases gradually in order to follow changes in building regulations.

In addition to three building envelope elements, two interior construction elements have to be defined: internal ceiling and internal partition. Internal ceiling was made from 100mm cast concrete (dense), while internal partition was composed of two gypsum plasterboard (each 25mm thick) separated by 10mm air gap.

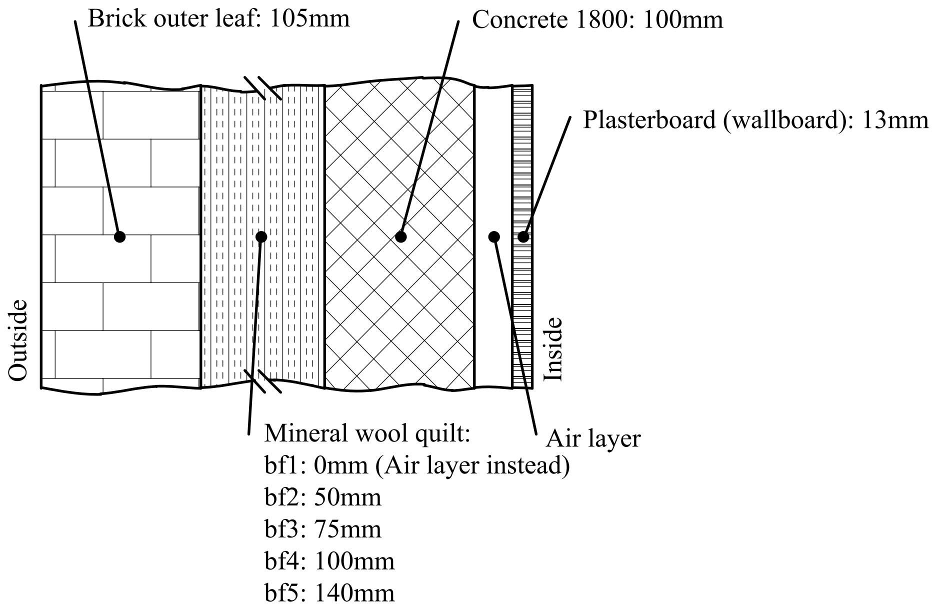


Figure A.3. External wall – cross section

The most important properties of materials used to define building elements, are presented in Table A.1 Property values were collected from ASHRAE Fundamentals Handbook (ASHRAE, 2009) and DesignBuilder database of material (DesignBuilder, v.2). DesignBuilder database of material is primarily based on data from various sources such as ASHRAE Fundamentals Handbook, DOE/EnergyPlus Software, UK NCM, CIBSE, etc. In addition, it also includes material properties obtained directly from manufacturers, for example from Uralita for insulating materials. Particularly large is the database of glazing properties which contains data from wide range of glazing companies such as AFG, Cardinal, Glaverbel, Pilkington, South Technologies, Vitro, etc.

Air layer in the external wall element and the air gap in the interior partition are assumed to be materials with no mass and they are defined by specifying thermal resistance only, which was set to 0.18 $m^2 \cdot K/W$ for the former and 0.15 $m^2 \cdot K/W$ for the later.

Table A.1. Material properties

Material	Conductivity [W/m·K]	Density [kg/m3]	Specific Heat [J/kg·K]
Material	0.7	2100	1000
Asphalt	0.77	1700	840
Brick outer leaf	0.77	1700	1000
Brick slips	1.35	2000	1000
Cast concrete	1.4	2100	840
Cast concrete - dense	1.5	1500	2085
Clay underfloor	1.93	2400	1000
Concrete - dense	1.13	1800	1000
Concrete 1800	0.04	15	1300
Expanded Polystyrene (EPS)	0.41	1200	1000
Flooring screed	0.25	900	1000
Gypsum Plasterboard	0.04	12	1030
Min wool quilt	0.036	50	1030
Mineral wool - dense	0.21	900	1000
Plasterboard (wallboard)	2	2000	1000
Stone chippings	0.7	2100	1000

Similar to opaque building elements, fenestration was also defined in EnergyPlus by the composition of several layers. Five different glazing units were created to match with corresponding building fabric type. Single-glazed unit was made from 4 mm clear glass pane and was used in combination with building fabric 1. Building fabric 2 and building fabric 3 glazing units were made from two clear glass panes. The difference between two units is in the air cavity depth which was set either to 6 mm in the case of building fabric 2 window or to 12 mm for the building fabric 3 window. Building fabric 4 and building fabric 5 windows are also double-glazed with the clear glass outer pane, while the inner pane was made of clear glass with the Low-E characteristics. In both cases, the cavity between two panes was 16 mm depth. However, the cavity was filed with an air in the building fabric 4 window, while the best practice glazing (building fabric 5) had xenon instead of air. One-half of all simulation scenarios were using windows with reflective characteristics, which was obtained by replacing outer clear glass pane with reflective glass pane. Cross section of all glazing units can be seen in Figure A.4.

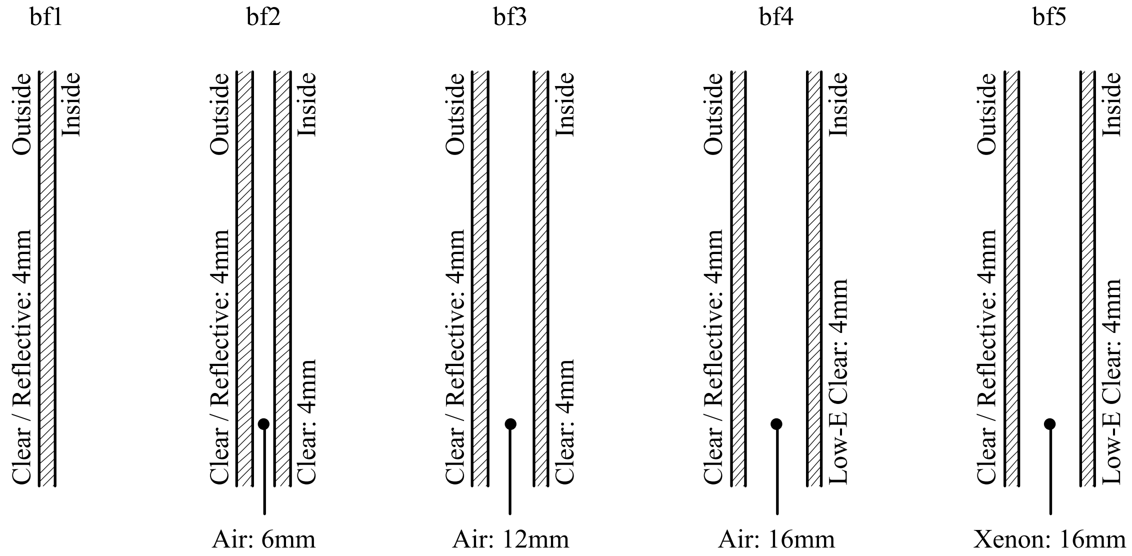


Figure A.4. Glazing – cross section

Properties of glazing panes used for windows in this research are presented in Table A.2. Terms "Front Side" and "Back Side" are used for the side of the layer opposite the zone in which the window is defined and the side closest to the zone respectively. This means that for exterior windows, "front side" is the side closest to the outdoors. Solar properties of the glazing are used for load calculation as the solar radiation transmitted by the window layers enters the zone and is a component of the

Table A.2. Glazing pane properties

Glazing Pane	Clear 4mm	Low-E Clear 4mm	Reflective 4mm
Solar Transmittance at Normal Incidence	0.816	0.62	0.429
Front Side Solar Reflectance at Normal Incidence	0.075	0.075	0.308
Back Side Solar Reflectance at Normal Incidence	0.075	0.075	0.379
Visible Transmittance at Normal Incidence	0.892	0.847	0.334
Front Side Visible Reflectance at Normal Incidence	0.081	0.081	0.453
Back Side Visible Reflectance at Normal Incidence	0.081	0.081	0.505
Infrared Transmittance at Normal Incidence	0	0	0
Front Side Infrared Hemispherical Emissivity	0.84	0.84	0.84
Back Side Infrared Hemispherical Emissivity	0.84	0.1	0.82
Conductivity [W/m·K]	1	1	1

Table A.3. Regular/Reflective glazing properties

		Clear / Reflective [mm]	Air (Xenon for BF5) [mm]	Clear [mm]	Low-E Clear [mm]	SHGC	Light Transmittance	U-value [W/m^2·K]
BF1	**Reg.**	4				0.847	0.892	5.87
	Ref.					0.505	0.335	5.81
BF2	**Reg.**	4	6	4		0.74	0.801	3.15
	Ref.					0.435	0.312	3.13
BF3	**Reg.**	4	12	4		0.742	0.801	2.73
	Ref.					0.432	0.312	2.71
BF4	**Reg.**	4	16		4	0.631	0.761	1.92
	Ref.					0.362	0.296	1.91
BF5	**Reg.**	4	16		4	0.637	0.761	1.78
	Ref.					0.365	0.296	1.77

zone load, while the visible transmittance and reflectance properties are used in the daylighting calculation.

Table A.3 presents layers thickness and the order in each of five different glazing units. Moreover, window solar heat gain coefficient (SHGC) and light transmittance values show the obvious difference between regular and reflective glazing. Last column shows overall U-values.

Appendix B. EnergyPlus Simulation Software

This appendix presents the detailed literature survey on validation of EnergyPlus building simulation software.

One of the biggest advantages of EnergyPlus, when compared to its predecessors BLAST and DOE-2, is simultaneous simulation of loads, systems and plants which results in better accuracy of simulation outputs. Namely, this integrated simulation approach allows system and plant output to directly affect building thermal response rather than to calculate building loads firstly and then simulate systems and plants with no feedback from one to the other, which was the case in two predecessor programs. Crawley et al. (2001a) described the concept of integrated simulation approach. For a user-specified time step, loads calculated by a heat balance engine are passed to the building system simulation module which calculates heating/cooling requirements of system and plant as well as electricity requirements. Inequality in building loads and building systems simulation module outputs, either if building loads are not met due to undersized equipment for example, or exceeded in cases when more energy than required is delivered by a system, results in the adjustments of a space temperature/humidity in the next time step of the load calculations. Integrated simulation also allows evaluation of more realistic control systems, moisture adsorption and desorption in building fabric elements, radiant heating and cooling systems, and interzone air flow (Crawley et al., 2008).

An important part of EnergyPlus development is testing and validating by using both analytical and comparative tests. Analytical tests compare the simulation outputs against mathematical solutions, while comparative tests compare against other software. Each EnergyPlus major release is tested using following tests:

Analytical tests:

- Building fabric tests, based on ASHRAE Research Project 1052,
- HVAC tests, based on ASHRAE Research Project 865,

Comparative tests:

- ANSI/ASHRAE Standard 140-2007,
- International Energy Agency Solar Heating and Cooling Programme (IEA SHC) BESTest (Building Energy Simulation Test),
- EnergyPlus HVAC Component Comparative tests,
- EnergyPlus Global Heat Balance tests.

ASHRAE Research Project 1052 developed analytical tests for the building fabric (Spitler et al., 2001). Tests cover a variety of building envelope mechanisms including convection, conduction, radiant transfer, solar gains, internal gains, air infiltration, shading, and ground coupling. EnergyPlus predictions compared very closely with the analytical results obtained from ASHRAE Research Project 1052 although there are certain differences, mainly in window heat gains, treatment of external long wave radiation and ground-coupled heat transfer for slabs (EnergyPlus, 2010c).

ANSI/ASHRAE Standard 140-2007 (ANSI/ASHRAE, 2007c) specifies test procedures for evaluating the technical capabilities and ranges of applicability of computer programs that calculate the thermal performance of buildings and their HVAC systems. The Standard is primarily based on the BESTEST suites developed through several International Energy Agency (IEA) projects, in particular Tasks 12 and 22. Part of the Standard which covers testing and diagnostic procedure for thermal models related to the architectural fabric of the building has its roots in the IEA Task 12 report *"Building Energy Simulation Test (BESTEST) and Diagnostic Methods"* (Judkoff and Neymark, 1995). Various tests specified in this section include buildings with both low mass and high mass construction, without windows and with windows on various exposures, with and without following characteristics: exterior window shading, temperature setback, night ventilation, and free floating space temperatures. Tests results predicted by EnergyPlus were compared to results from eight other whole building energy simulation programs and EnergyPlus was within the range of spread of results for the other eight programs for almost all modelled scenarios which amounted over 100 (EnergyPlus, 2010d).

HVAC BESTEST Cases E100-E200 (Neymark and Judkoff, 2002) and E300-E545 (Neymark and Judkoff, 2004), originally developed through IEA Task 22 and later incorporated into ANSI/ASHRAE Standard 140-2007, specify procedures for testing the ability of whole-building simulation programs to model the performance of unitary space cooling equipment. Various combination of parameters allow the wide range of operating conditions to be analysed, such as dry coil versus wet coil operation or the influence of part-loading of equipment, all in order to test the equipment in various domains of the performance map. Additional set of test cases evaluate program"s modelling capabilities but in hourly dynamic context. Output values which are compared among several program"s and in the first set of test cases against three analytical solutions are: compressor and fan electricity consumption, cooling coil sensible and latent loads, coefficient of performance (COP), and zone conditions such as temperature and humidity ratio. EnergyPlus was tested using both test cases sets. Results, when compared to the three analytical solutions of steady-state cases E100-E200, generally agreed within 1.1% except for the prediction of the mean zone humidity ratio for high sensible heat ratio cases which agreed within 2.7% (EnergyPlus, 2010e). Various dynamic test cases (E300-E545) showed that total energy consumption predicted by six software packages differed by 2% - 6% compared to the mean. EnergyPlus results followed that trend and generally fell within min/max of the results for each case (EnergyPlus, 2010f).

IEA Task 34 (IEA, 2007) developed several new procedures for testing and validating building simulation programs. These procedures represent an extension to previously described BESTEST suites. Although not yet implemented in ANSI/ASHRAE Standard 140, two of them are of particular interest as they were used for EnergyPlus validation:

- Multi-zone non-airflow in-depth diagnostic cases (Neymark and Judkoff, 2008), and
- Mechanical equipment and control strategies for a chilled water and a hot water system (Felsmann, 2008).

Multi-zone non-airflow diagnostic cases specified five test cases, which included multi-zone with no windows, multi-zone with unshaded windows, multi-zone with shaded windows for two configurations, and multi-zones in series with internal windows, in order to test prediction of zone loads and resulting zone temperatures. EnergyPlus results were compared to an analytical solution for the first case, multi-zone with no windows, and predicted sensible cooling load agreed exactly with the analytical solution. Results for latter four cases, which were compared against five other programs, showed good prediction as they were within the bounds of the other building simulation programs results (EnergyPlus, 2010h).

Chilled water cooling coil and hot water heating coil tests specified in IEA Mechanical Equipment & Control Strategies for a Chilled Water and a Hot Water System (Felsmann, 2008) were used to test the ability of EnergyPlus to calculate coil loads and leaving water temperature. Two types of water side coil control were analysed: variable water flow rate with a constant water inlet temperature and constant water flow rate with a variable water inlet temperature. EnergyPlus results were compared to the results from four other programs. Cooling coil tests showed reasonable accuracy as the total, sensible and latent annual cooling loads predicted by EnergyPlus were within the range of results of other programs. Similar conclusion was driven from heating coil tests where the results for total heating loads varied by less than 5% between all five programs (EnergyPlus, 2010g).

Besides analytical and comparative testing, EnergyPlus, or at least some of its modules, was also validated empirically. As briefly mentioned in Chapter 2, Maamari et al. (2006) compared several daylight simulation tools in empirical validation study, including DElight which is a part of EnergyPlus. They assessed the capability of four lighting simulation methods to predict the performance of Complex Fenestration Systems (CFS). Measurements were conducted under both artificial sky and real sky conditions. The comparison between experimental data and the simulation results showed that illuminance can be predicted with acceptable accuracy, generally within the range of 10% to 20%.

Loutzenhiser et al. (2006, 2009) investigated capabilities of four building simulation programs in respect to simulating energy flow through windows. Impact of solar heat gains and associated interactions were evaluated by inspecting the cooling power required to maintain constant air temperature within the test cell. Predictions of a solar power transmitted through the window were similar for all tested programs although EnergyPlus performed slightly better than other tools.

The same team also did an empirical validation of modelling solar gain through glazing unit with external and internal shading screen (Loutzenhiser et al., 2007b). Two experiments, one with external and another with internal diffuse window shading screen, were run for a 20-day periods to assess the performance of four building simulation programs. EnergyPlus results for the external shading screen experiment were validated within 95% credible limits. The absolute average differences between measurements and EnergyPlus predictions were below 3.7%. Although showing the best performance with external shading screen, when compared to other programs, EnergyPlus was less accurate in internal shading screen experiment where the absolute average differences were 6.7% with predictions.

In another experiment Loutzenhiser et al. (2007a) analysed the accuracy of seven solar radiation models implemented in four building energy simulation programs, including 1990 Perez model integrated in EnergyPlus. Analysis showed that simulation programs are capable to precisely calculate total solar energy irradiated on building facades but only for a longer time periods. On the other hand, they are limited in predictions of solar irradiance at specific point in time as the difference between measurement and programs predictions at specific time step were even above 100 W/m^2. This inaccuracy at specific time step can result in incorrect cooling/heating loads predictions which can affect other features of building simulation programs such as sizing of HVAC equipment, control of HVAC system or sizing of shading devices.

Chantrasrisalai et al. (2003) evaluated the EnergyPlus low-temperature radiant model by comparing its predicted total energy consumption and operative temperature against measured data. The measured data were obtained from a previous study done by Scheatzle (2003). Monitored building was residential with high mass walls, insulated externally, and radiant panels in both the ceiling and the floor. Ceiling panels were used for cooling and floor panels were used for heating. Chantrasrisalai et al. simulated only the cooling side of the system and found that simulated results agreed well with the measured data for both cooling energy consumptions and indoor operative temperatures.

Overall conclusion, which can be driven form abovementioned studies, is that EnergyPlus simulation software is very well validated by using both comparative tests and analytical tests, as well as against experimental results. It can be used to predict building thermal performance and performance of HVAC systems with a high accuracy.

Appendix C. HVAC Systems Regression Models

This appendix consists of tables and figures which were generated by the regression analysis. The complete regression analysis procedure, presented in Chapter 4, was based on the VAV system. All other systems (CAV, FC, EMB and ALU) were described in Chapter 4 by summary tables. Their complete analysis outputs are presented here through sets of detailed tables and charts.

C.1. Variable air volume system (VAV)

- **Cooling energy requirements**

Table C.1. Regression model parameter values, standard errors and 95% confidence bounds for models of VAV system cooling energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	6.843×10^{-1}	2.225×10^{-3}	6.799×10^{-1} / 6.886×10^{-1}
LIN2	a	6.065	4.362×10^{-2}	5.980 / 6.151
	b	4.807×10^{-1}	1.722×10^{-3}	$4.773 \times$ 10-1 / 4.840×10^{-1}
QUAD	a	4.268	6.668×10^{-2}	4.137 / 4.399
	b	6.498×10^{-1}	5.344×10^{-3}	6.393×10^{-1} / 6.603×10^{-1}
	c	-2.879×10^{-3}	8.719×10^{-5}	-3.050×10^{-3} / -2.708×10^{-3}
POW	a	1.449	1.934×10^{-1}	1.070 / 1.828
	b	1.829	7.301×10^{-2}	1.686 / 1.972
	c	6.972×10^{-1}	8.697×10^{-3}	6.802×10^{-1} / 7.143×10^{-1}
POLY11	a	3.764	7.277×10^{-2}	3.621 / 3.907
	b	5.139×10^{-1}	1.733×10^{-3}	5.106×10^{-1} / 5.173×10^{-1}
	c	3.239×10^{-2}	8.779×10^{-4}	3.067×10^{-2} / 3.411×10^{-2}
POLY21	a	3.696	5.998×10^{-2}	3.578 / 3.813
	b	5.128×10^{-1}	3.976×10^{-3}	5.050×10^{-1} / 5.206×10^{-1}
	c	-5.272×10^{-3}	7.279×10^{-4}	-6.699×10^{-3} / -3.845×10^{-3}
	d	-1.839×10^{-3}	4.575×10^{-5}	-1.929×10^{-3} / -1.749×10^{-3}
	e	3.574×10^{-3}	4.420×10^{-5}	3.488×10^{-3} / 3.661×10^{-3}
POLY12	a	4.875	9.259×10^{-2}	4.693 / 5.056
	b	3.732×10^{-1}	1.884×10^{-3}	3.695×10^{-1} / 3.769×10^{-1}
	c	5.441×10^{-3}	3.053×10^{-3}	-5.435×10^{-4} / 1.143×10^{-2}
	e	4.461×10^{-3}	4.744×10^{-5}	4.368×10^{-3} / 4.554×10^{-3}
	f	-2.096×10^{-4}	2.042×10^{-5}	-2.496×10^{-4} / -1.695×10^{-4}
POLY22	a	1.069	1.009×10^{-1}	8.706×10^{-1} / 1.266
	b	5.792×10^{-1}	4.164×10^{-3}	5.710×10^{-1} / 5.874×10^{-1}
	c	7.421×10^{-2}	2.666×10^{-3}	6.899×10^{-2} / 7.944×10^{-2}
	d	-2.299×10^{-3}	4.363×10^{-5}	-2.385×10^{-3} / -2.214×10^{-3}
	e	2.628×10^{-3}	5.015×10^{-5}	2.529×10^{-3} / 2.726×10^{-3}
	f	-5.091×10^{-4}	1.656×10^{-5}	-5.415×10^{-4} / -4.766×10^{-4}

Table C.2. Comparison of relative differences between predicted and observed values for models of VAV system cooling energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	17.03	-1.66	-0.78	-0.63	-1.70	-0.21	-0.49	-0.21
Std. Dev.	19.896	10.464	8.220	7.984	10.592	3.269	4.558	3.089
Maximum	70.89	23.59	21.10	20.93	16.54	10.29	12.41	12.49
Minimum	-34.43	-59.13	-29.77	-22.46	-87.29	-15.22	-34.18	-20.21
Perc. 25	3.12	-7.12	-6.42	-6.37	-4.74	-1.97	-2.30	-1.75
Perc. 75	31.43	5.19	5.10	5.31	4.52	1.87	2.29	1.72

- ## Heating energy requirements

Table C.3. Regression model parameter values, standard errors and 95% confidence bounds for models of VAV system heating energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	8.908×10^{-1}	1.097×10^{-3}	8.887×10^{-1} / 8.930×10^{-1}
LIN2	a	-1.683	1.257×10^{-1}	-1.929 / -1.436
	b	9.175×10^{-1}	2.263×10^{-3}	9.131×10^{-1} / 9.219×10^{-1}
QUAD	a	-1.222	3.570×10^{-1}	-1.922 / -5.219×10^{-1}
	b	8.988×10^{-1}	1.377×10^{-2}	8.718×10^{-1} / 9.258×10^{-1}
	c	1.477×10^{-4}	1.071×10^{-4}	-6.229×10^{-5} / 3.577×10^{-4}
POW	a	-9.102×10^{-1}	6.095×10^{-1}	-2.105 / 2.847×10^{-1}
	b	8.442×10^{-1}	5.522×10^{-2}	7.359×10^{-1} / 9.524×10^{-1}
	c	1.016	1.288×10^{-2}	9.911×10^{-1} / 1.042
POLY11	a	-6.516	1.981×10^{-1}	-6.904 / -6.128
	b	1.403×10^{-1}	4.717×10^{-3}	1.311×10^{-1} / 1.496×10^{-1}
	c	9.545×10^{-1}	2.390×10^{-3}	9.498×10^{-1} / 9.592×10^{-1}
POLY21	a	5.952×10^{-1}	3.907×10^{-1}	-1.707×10^{-1} / 1.361
	b	-3.901×10^{-1}	2.589×10^{-2}	-4.409×10^{-1} / -3.394×10^{-1}
	c	8.535×10^{-1}	4.741×10^{-3}	8.442×10^{-1} / 8.628×10^{-1}
	d	4.909×10^{-3}	2.980×10^{-4}	4.325×10^{-3} / 5.493×10^{-3}
	e	7.008×10^{-3}	2.879×10^{-4}	6.444×10^{-3} / 7.573×10^{-3}
POLY12	a	-3.811	5.302×10^{-1}	-4.851 / -2.772
	b	-3.675×10^{-4}	1.079×10^{-2}	-2.152×10^{-2} / 2.079×10^{-2}
	c	8.693×10^{-1}	1.748×10^{-2}	8.350×10^{-1} / 9.036×10^{-1}
	e	4.251×10^{-3}	2.717×10^{-4}	3.718×10^{-3} / 4.783×10^{-3}
	f	2.560×10^{-4}	1.169×10^{-4}	2.679×10^{-5} / 4.853×10^{-4}
POLY22	a	5.834	7.272×10^{-1}	4.408 / 7.259
	b	-5.225×10^{-1}	3.000×10^{-2}	-5.813×10^{-1} / -4.636×10^{-1}
	c	6.950×10^{-1}	1.921×10^{-2}	6.574×10^{-1} / 7.327×10^{-1}
	d	5.827×10^{-3}	3.143×10^{-4}	5.211×10^{-3} / 6.443×10^{-3}
	e	8.896×10^{-3}	3.613×10^{-4}	8.187×10^{-3} / 9.604×10^{-3}
	f	1.015×10^{-3}	1.193×10^{-4}	7.812×10^{-4} / 1.249×10^{-3}

Table C.4. Comparison of relative differences between predicted and observed values for models of VAV system heating energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	-2.88	-0.90	-0.95	-0.94	-1.02	-0.51	-0.79	-0.68
Std. Dev.	9.479	8.865	8.886	8.885	8.988	7.992	8.359	7.836
Maximum	42.07	45.04	44.75	44.74	30.74	30.75	35.74	31.13
Minimum	-24.63	-21.83	-21.53	-21.56	-35.96	-39.79	-23.56	-41.47
Perc. 25	-8.41	-6.01	-5.99	-5.99	-5.42	-5.59	-5.93	-5.30
Perc. 75	0.48	1.16	1.18	1.17	4.28	3.63	4.07	2.84

- **Auxiliary energy requirements**

Table C.5. Regression model parameter values, standard errors and 95% confidence bounds for models of VAV system auxiliary energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
POLY11	a	3.436×10^{-2}	9.720×10^{-2}	-1.562×10^{-1} / 2.249×10^{-1}
	b	3.779×10^{-1}	2.315×10^{-3}	3.734×10^{-1} / 3.824×10^{-1}
	c	1.087×10^{-1}	1.173×10^{-3}	1.064×10^{-1} / 1.110×10^{-1}
POLY21	a	1.586	1.128×10^{-1}	1.364 / 1.807
	b	2.556×10^{-1}	7.479×10^{-3}	2.409×10^{-1} / 2.702×10^{-1}
	c	4.554×10^{-2}	1.369×10^{-3}	4.286×10^{-2} / 4.823×10^{-2}
	d	-8.386×10^{-4}	8.607×10^{-5}	-1.007×10^{-3} / -6.698×10^{-4}
	e	5.409×10^{-3}	8.315×10^{-5}	5.246×10^{-3} / 5.572×10^{-3}
POLY12	a	2.709	1.499×10^{-1}	2.415 / 3.003
	b	1.840×10^{-1}	3.050×10^{-3}	1.780×10^{-1} / 1.900×10^{-1}
	c	2.979×10^{-2}	4.942×10^{-3}	2.010×10^{-2} / 3.948×10^{-2}
	e	5.995×10^{-3}	7.680×10^{-5}	5.844×10^{-3} / 6.146×10^{-3}
	f	4.549×10^{-5}	3.305×10^{-5}	-1.932×10^{-5} / 1.103×10^{-4}
POLY22	a	1.213	2.119×10^{-1}	7.972×10^{-1} / 1.628
	b	2.650×10^{-1}	8.742×10^{-3}	2.478×10^{-1} / 2.821×10^{-1}
	c	5.682×10^{-2}	5.597×10^{-3}	4.585×10^{-2} / 6.780×10^{-2}
	d	-9.039×10^{-4}	9.160×10^{-5}	-1.083×10^{-3} / -7.243×10^{-4}
	e	5.274×10^{-3}	1.053×10^{-4}	5.068×10^{-3} / 5.481×10^{-3}
	f	-7.225×10^{-5}	3.476×10^{-5}	-1.404×10^{-4} / -4.105×10^{-6}

Table C.6. Comparison of relative differences between predicted and observed values for models of VAV system auxiliary energy consumption in office buildings

	POLY11	POLY21	POLY12	POLY22
Mean	-1.46	-0.43	-0.55	-0.42
Std. Dev.	11.491	5.982	6.141	5.982
Maximum	27.88	20.24	20.07	19.93
Minimum	-48.41	-18.12	-19.02	-17.75
Perc. 25	-6.47	-4.61	-4.84	-4.59
Perc. 75	5.65	3.65	3.69	3.65

C.2. Constant air volume system (CAV)

- **Cooling energy requirements**

Table C.7. Regression model parameter values, standard errors and 95% confidence bounds for models of CAV system cooling energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	5.319×10^{-1}	1.714×10^{-3}	5.285×10^{-1} / 5.353×10^{-1}
LIN2	a	4.363	4.308×10^{-2}	4.278 / 4.447
	b	3.854×10^{-1}	1.700×10^{-3}	3.821×10^{-1} / 3.888×10^{-1}
QUAD	a	2.913	6.890×10^{-2}	2.778 / 3.048
	b	5.219×10^{-1}	5.523×10^{-3}	5.111×10^{-1} / 5.327×10^{-1}
	c	-2.322×10^{-3}	9.010×10^{-5}	-2.499×10^{-3} / -2.146×10^{-3}
POW	a	7.539×10^{-1}	1.982×10^{-1}	3.653×10^{-1} / 1.143
	b	1.431	7.396×10^{-2}	1.286 / 1.576
	c	7.026×10^{-1}	1.128×10^{-2}	6.804×10^{-1} / 7.247×10^{-1}
POLY11	a	2.249	7.357×10^{-2}	2.105 / 2.393
	b	4.160×10^{-1}	1.752×10^{-3}	4.126×10^{-1} / 4.195×10^{-1}
	c	2.975×10^{-2}	8.876×10^{-4}	2.801×10^{-2} / 3.149×10^{-2}
POLY21	a	2.417	9.781×10^{-2}	2.225 / 2.609
	b	3.983×10^{-1}	6.483×10^{-3}	3.856×10^{-1} / 4.110×10^{-1}
	c	-4.938×10^{-3}	1.187×10^{-3}	-7.265×10^{-3} / -2.611×10^{-3}
	d	-1.383×10^{-3}	7.461×10^{-5}	-1.529×10^{-3} / -1.236×10^{-3}
	e	3.211×10^{-3}	7.208×10^{-5}	3.070×10^{-3} / 3.352×10^{-3}
POLY12	a	4.253	1.339×10^{-1}	3.991 / 4.516
	b	2.805×10^{-1}	2.725×10^{-3}	2.752×10^{-1} / 2.858×10^{-1}
	c	-3.035×10^{-2}	4.415×10^{-3}	-3.900×10^{-2} / -2.169×10^{-2}
	e	4.172×10^{-3}	6.861×10^{-5}	4.038×10^{-3} / 4.307×10^{-3}
	f	7.113×10^{-5}	2.953×10^{-5}	1.324×10^{-5} / 1.290×10^{-4}
POLY22	a	1.78	1.834×10^{-1}	1.420 / 2.139
	b	4.144×10^{-1}	7.566×10^{-3}	3.996×10^{-1} / 4.292×10^{-1}
	c	1.434×10^{-2}	4.844×10^{-3}	4.847×10^{-3} / 2.384×10^{-2}
	d	-1.494×10^{-3}	7.927×10^{-5}	-1.650×10^{-3} / -1.339×10^{-3}
	e	2.981×10^{-3}	9.112×10^{-5}	2.803×10^{-3} / 3.160×10^{-3}
	f	-1.235×10^{-4}	3.008×10^{-5}	-1.825×10^{-4} / -6.452×10^{-5}

Table C.8. Comparison of models of CAV system cooling energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	12.67	11.46	12.67	12.67	12.67	12.67	12.67	12.67	12.67
σ_y	5.325	7.089	5.137	5.165	5.165	5.180	5.269	5.264	5.269
RSS		27787	7568	6451	6463	5854	2304	2507	2294
R^2		0.7448	0.9305	0.9408	0.9406	0.9462	0.9788	0.977	0.9789
RMSD		2.69	1.4039	1.2961	1.2973	1.2347	0.7746	0.808	0.7729
e_{max}		7.35	7.00	6.45	6.57	5.54	2.93	3.04	2.86
e_{min}		-10.89	-4.71	-2.78	-2.57	-5.45	-3.22	-3.66	-3.08
$\overline{\lvert e\rvert}$		2.22	1.05	0.98	0.99	0.90	0.59	0.62	0.59
$\lvert e\rvert_{95\%}$		5.15	2.90	2.49	2.47	2.57	1.62	1.73	1.61

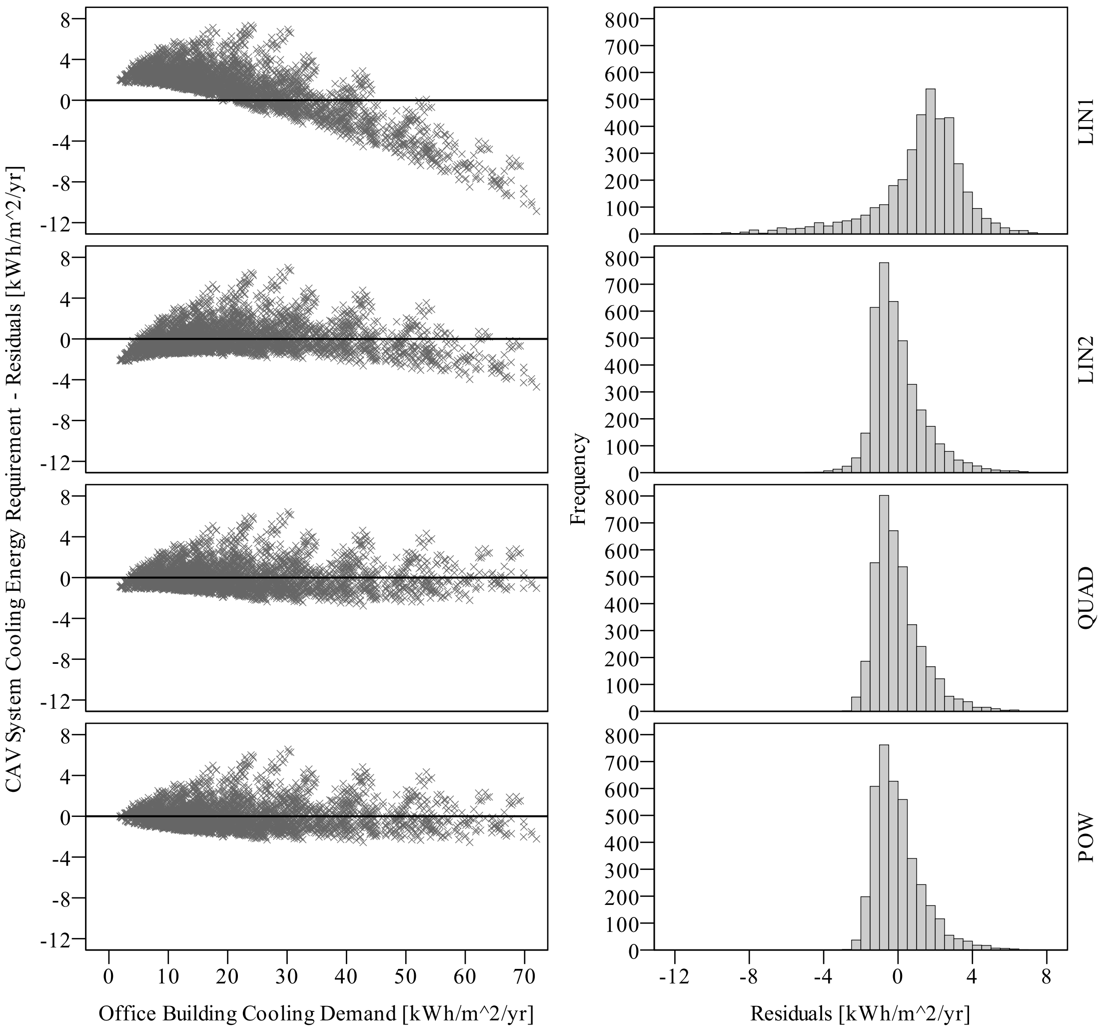


Figure C.1. Residuals scatter plots and histograms for single independent variable models of CAV system cooling energy consumption in office buildings

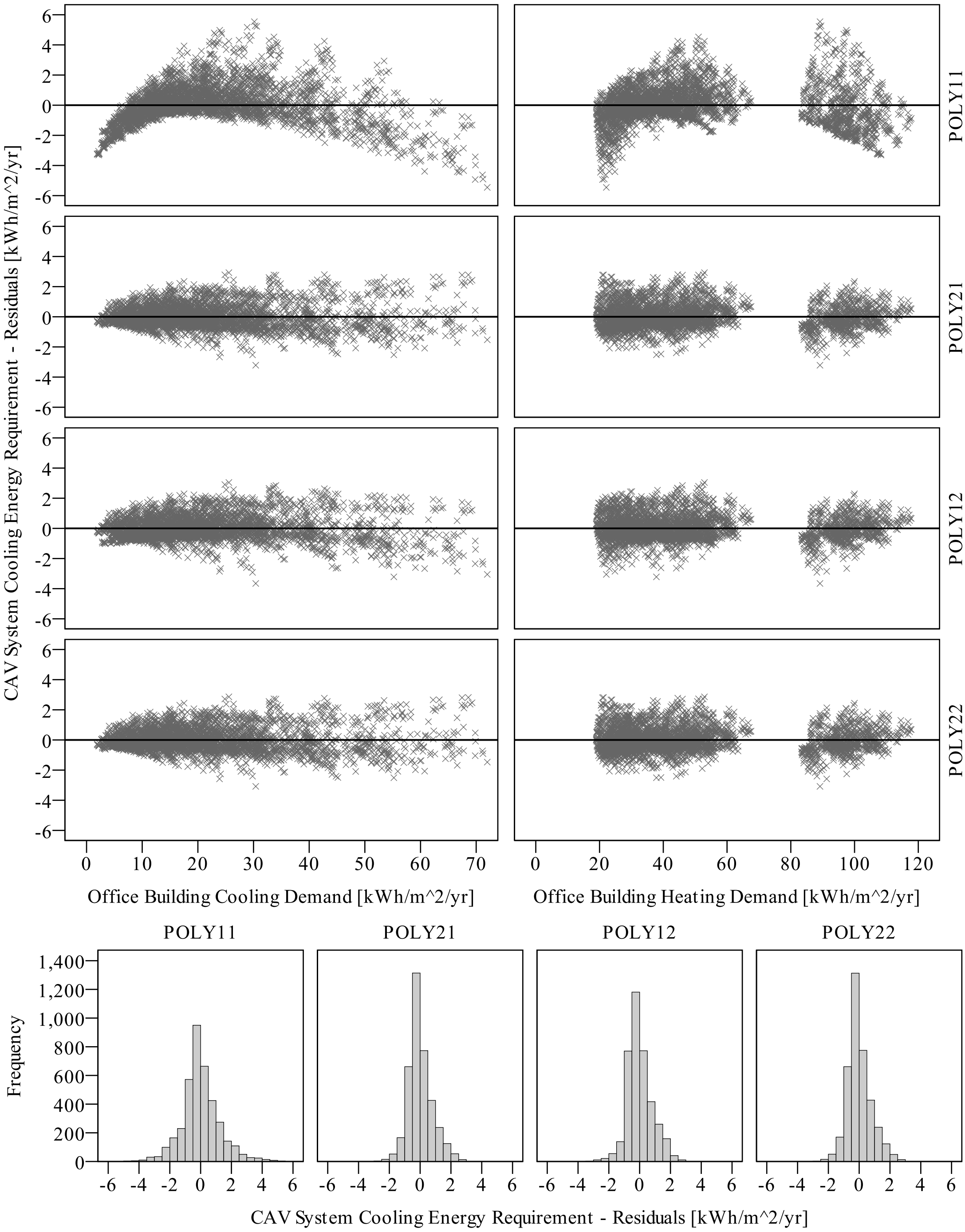


Figure C.2. Residuals scatter plots and histograms for two independent variables models of CAV system cooling energy consumption in office buildings

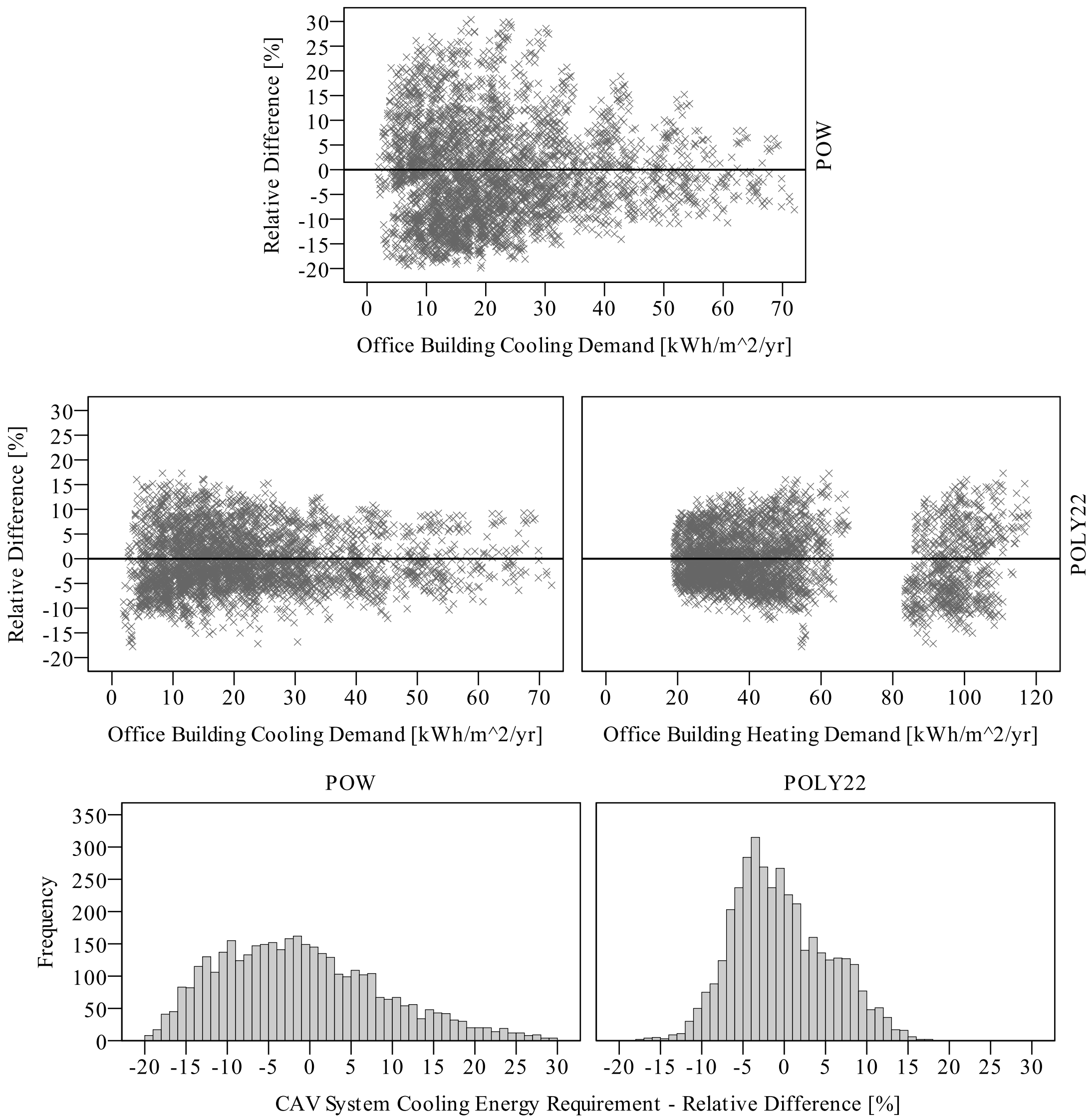


Figure C.3. Residuals scatter plots and histograms for POW and POLY22 models of CAV system cooling energy consumption in office buildings

Table C.9. Comparison of relative differences between predicted and observed values for models of CAV system cooling energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	15.85	-2.13	-1.14	-0.99	-2.18	-0.40	-0.65	-0.41
Std. Dev.	19.502	12.092	9.990	9.856	12.543	5.765	6.395	5.772
Maximum	66.48	32.40	31.02	30.37	24.77	18.13	17.68	17.30
Minimum	-39.74	-71.86	-33.82	-19.88	-110.11	-17.76	-29.97	-17.78
Perc. 25	2.57	-8.98	-8.63	-8.63	-5.99	-4.60	-4.94	-4.61
Perc. 75	29.85	5.01	5.12	5.27	4.75	3.53	3.69	3.53

- **Heating energy requirements**

Table C.10. Regression model parameter values, standard errors and 95% confidence bounds for models of CAV system heating energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	7.574×10^{-1}	1.304×10^{-3}	7.548×10^{-1} / 7.599×10^{-1}
LIN2	a	-5.901	1.196×10^{-1}	-6.136 / -5.666
	b	8.510×10^{-1}	2.154×10^{-3}	8.467×10^{-1} / 8.552×10^{-1}
QUAD	a	-3.869	3.380×10^{-1}	-4.531 / -3.206
	b	7.683×10^{-1}	1.304×10^{-2}	7.428×10^{-1} / 7.939×10^{-1}
	c	6.515×10^{-4}	1.014×10^{-4}	4.527×10^{-4} / 8.504×10^{-4}
POW	a	-2.625	5.191×10^{-1}	-3.643 / -1.607
	b	5.679×10^{-1}	3.829×10^{-2}	4.929×10^{-1} / 6.430×10^{-1}
	c	1.08	1.340×10^{-2}	1.054 / 1.106
POLY11	a	2.155	1.364×10^{-1}	1.888 / 2.423
	b	-2.339×10^{-1}	3.249×10^{-3}	-2.402×10^{-1} / -2.275×10^{-1}
	c	7.893×10^{-1}	1.646×10^{-3}	7.860×10^{-1} / 7.925×10^{-1}
POLY21	a	8.789×10^{-1}	1.616×10^{-1}	5.622×10^{-1} / 1.196
	b	-1.289×10^{-1}	1.071×10^{-2}	-1.499×10^{-1} / -1.079×10^{-1}
	c	8.680×10^{-1}	1.961×10^{-3}	8.641×10^{-1} / 8.718×10^{-1}
	d	1.929×10^{-3}	1.232×10^{-4}	1.688×10^{-3} / 2.171×10^{-3}
	e	-6.969×10^{-3}	1.190×10^{-4}	-7.202×10^{-3} / -6.736×10^{-3}
POLY12	a	4.881×10^{-1}	2.165×10^{-1}	6.368×10^{-2} / 9.126×10^{-1}
	b	6.217×10^{-3}	4.405×10^{-3}	-2.420×10^{-3} / 1.485×10^{-2}
	c	8.269×10^{-1}	7.138×10^{-3}	8.129×10^{-1} / 8.409×10^{-1}
	e	-7.636×10^{-3}	1.109×10^{-4}	-7.854×10^{-3} / -7.419×10^{-3}
	f	4.235×10^{-4}	4.774×10^{-5}	3.299×10^{-4} / 5.172×10^{-4}
POLY22	a	4.827	2.940×10^{-1}	4.250 / 5.403
	b	-2.286×10^{-1}	1.213×10^{-2}	-2.524×10^{-1} / -2.049×10^{-1}
	c	7.485×10^{-1}	7.767×10^{-3}	7.333×10^{-1} / 7.638×10^{-1}
	d	2.621×10^{-3}	1.271×10^{-4}	2.372×10^{-3} / 2.870×10^{-3}
	e	-5.547×10^{-3}	1.461×10^{-4}	-5.833×10^{-3} / -5.260×10^{-3}
	f	7.650×10^{-4}	4.823×10^{-5}	6.704×10^{-4} / 8.595×10^{-4}

Table C.11. Comparison of models of CAV system heating energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	35.72	37.04	35.72	35.72	35.72	35.72	35.72	35.72	35.72
σ_y	22.658	19.922	22.384	22.387	22.387	22.542	22.622	22.620	22.624
RSS		77314	47316	46813	46877	20130	6285	6553	5898
R^2		0.9608	0.976	0.9762	0.9762	0.9898	0.9968	0.9967	0.997
RMSD		4.4871	3.5103	3.4915	3.4939	2.2896	1.2793	1.3063	1.2393
e_{max}		10.34	6.85	7.01	6.98	6.99	4.07	4.07	3.64
e_{min}		-10.48	-13.17	-13.21	-13.25	-8.77	-4.46	-5.28	-5.40
$\overline{\lvert e \rvert}$		3.65	2.91	2.90	2.90	1.80	1.02	1.03	0.98
$\lvert e \rvert_{95\%}$		8.56	6.36	6.26	6.25	4.5	2.49	2.51	2.37

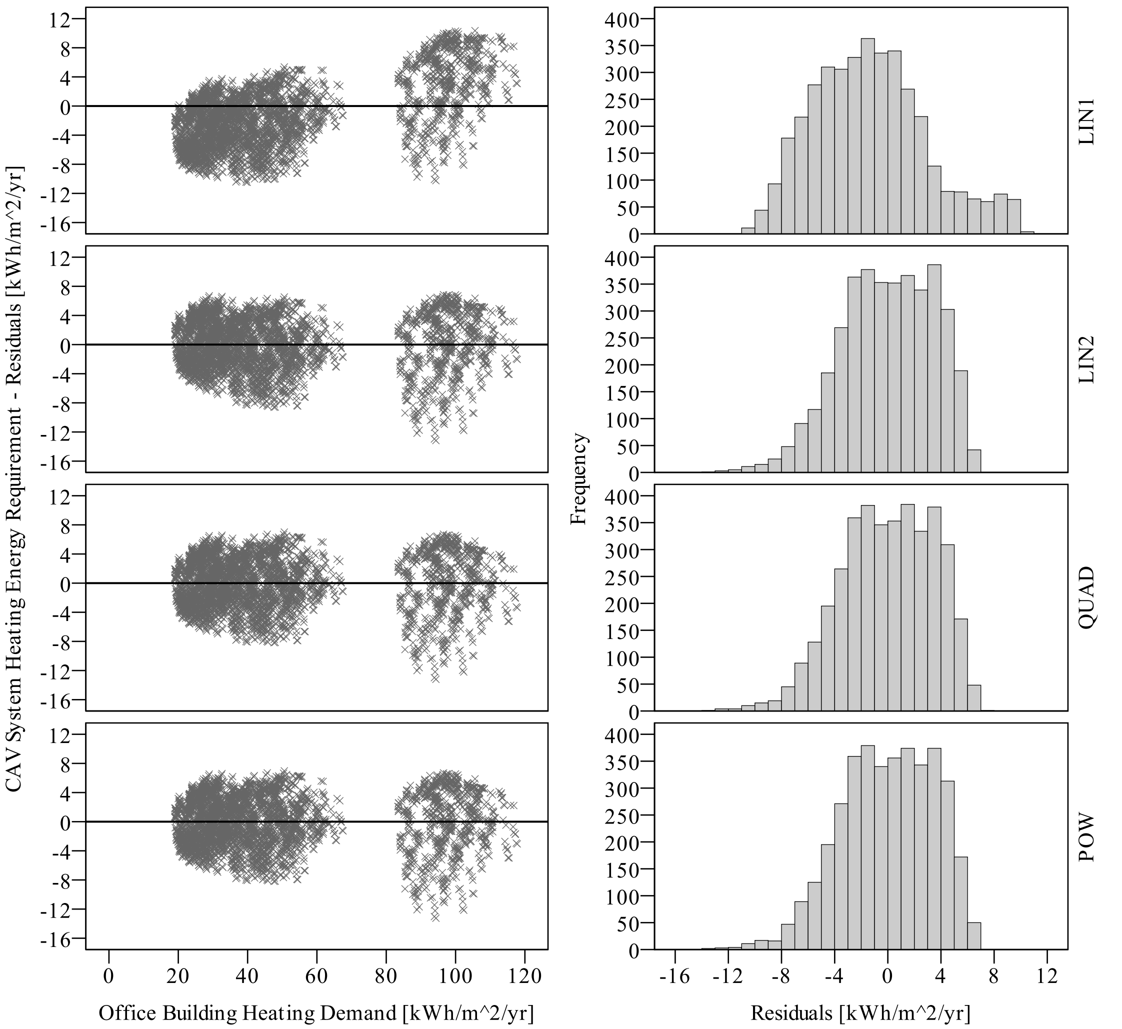


Figure C.4. Residuals scatter plots and histograms for single independent variable models of CAV system heating energy consumption in office buildings

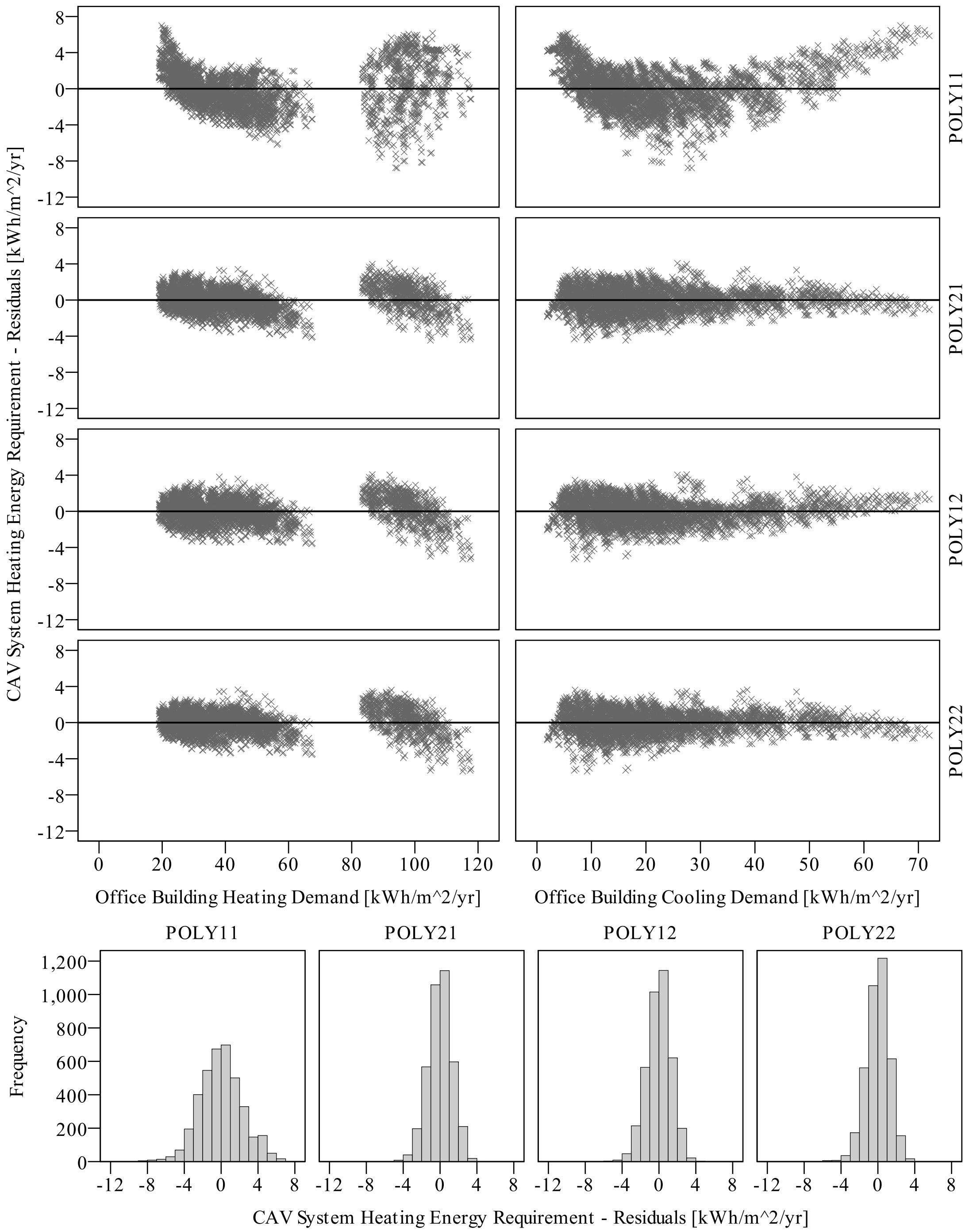


Figure C.5. Residuals scatter plots and histograms for two independent variables models of CAV system heating energy consumption in office buildings

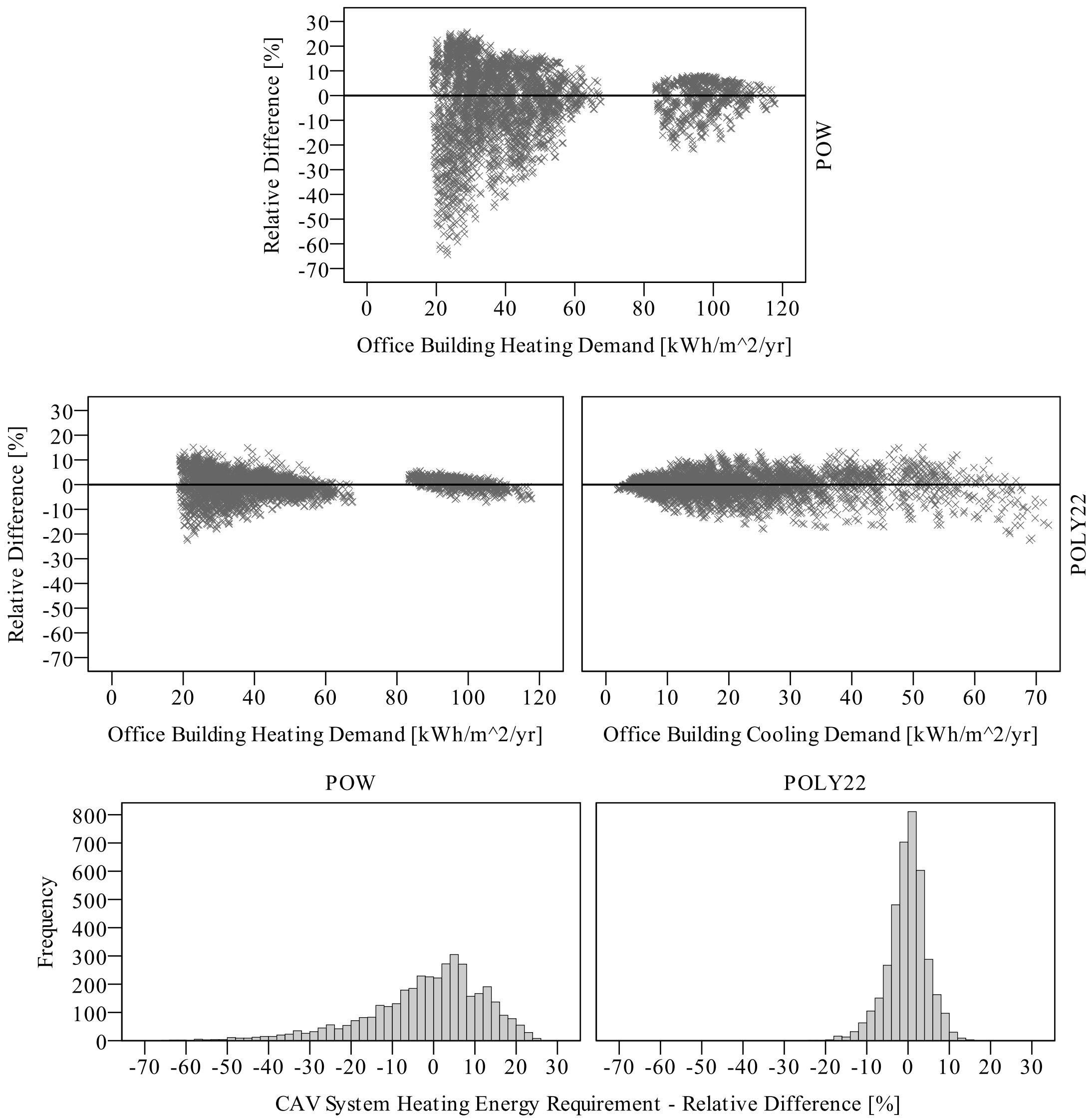


Figure C.6. Residuals scatter plots and histograms for POW and POLY22 models of CAV system heating energy consumption in office buildings

Table C.12. Comparison of relative differences between predicted and observed values for models of CAV system heating energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	-11.13	-1.59	-1.93	-1.93	1.13	-0.03	-0.13	-0.16
Std. Dev.	19.982	14.106	14.404	14.416	10.861	4.916	5.164	4.676
Maximum	13.79	27.11	25.62	25.71	76.06	16.38	20.77	15.10
Minimum	-104.31	-59.02	-64.31	-64.44	-19.66	-19.32	-22.55	-22.51
Perc. 25	-19.74	-9.08	-9.13	-9.10	-4.96	-2.76	-2.81	-2.57
Perc. 75	3.44	7.60	7.61	7.65	4.38	2.67	2.69	2.61

- **Auxiliary energy requirements**

Table C.13. Regression model parameter values, standard errors and 95% confidence bounds for models of CAV system auxiliary energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
POLY11	a	-7.978×10^{-2}	2.105×10^{-1}	-4.925×10^{-1} / 3.329×10^{-1}
	b	7.346×10^{-1}	5.012×10^{-3}	7.248×10^{-1} / 7.445×10^{-1}
	c	2.190×10^{-1}	2.539×10^{-3}	2.140×10^{-1} / 2.239×10^{-1}
POLY21	a	3.56	2.534×10^{-1}	3.063 / 4.056
	b	4.493×10^{-1}	1.680×10^{-2}	4.164×10^{-1} / 4.822×10^{-1}
	c	8.094×10^{-2}	3.076×10^{-3}	7.491×10^{-2} / 8.697×10^{-2}
	d	-1.493×10^{-3}	1.933×10^{-4}	-1.872×10^{-3} / -1.114×10^{-3}
	e	1.173×10^{-2}	1.868×10^{-4}	1.136×10^{-2} / 1.209×10^{-2}
POLY12	a	5.356	3.352×10^{-1}	4.699 / 6.013
	b	3.246×10^{-1}	6.820×10^{-3}	3.112×10^{-1} / 3.380×10^{-1}
	c	6.007×10^{-2}	1.105×10^{-2}	3.840×10^{-2} / 8.174×10^{-2}
	e	1.271×10^{-2}	1.717×10^{-4}	1.237×10^{-2} / 1.304×10^{-2}
	f	3.194×10^{-5}	7.392×10^{-5}	-1.130×10^{-4} / 1.769×10^{-4}
POLY22	a	2.609	4.758×10^{-1}	1.676 / 3.541
	b	4.733×10^{-1}	1.963×10^{-2}	4.348×10^{-1} / 5.118×10^{-1}
	c	1.097×10^{-1}	1.257×10^{-2}	8.507×10^{-2} / 1.344×10^{-1}
	d	-1.660×10^{-3}	2.057×10^{-4}	-2.063×10^{-3} / -1.257×10^{-3}
	e	1.138×10^{-2}	2.364×10^{-4}	1.092×10^{-2} / 1.185×10^{-2}
	f	-1.843×10^{-4}	7.805×10^{-5}	-3.373×10^{-4} / -3.125×10^{-5}

Table C.14. Comparison of models of CAV system auxiliary energy consumption in office buildings

	Observed	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	26.46	26.46	26.46	26.46	26.46
σ_y	9.099	8.385	8.875	8.872	8.875
RSS		47915	15468	15708	15445
R^2		0.8493	0.9513	0.9506	0.9514
RMSD		3.5324	2.007	2.0225	2.0055
e_{max}		16.15	8.34	8.43	8.41
e_{min}		-12.42	-8.09	-8.39	-7.89
$\overline{\lvert e \rvert}$		2.55	1.55	1.56	1.55
$\lvert e \rvert_{95\%}$		7.41	4.04	4.13	4.01

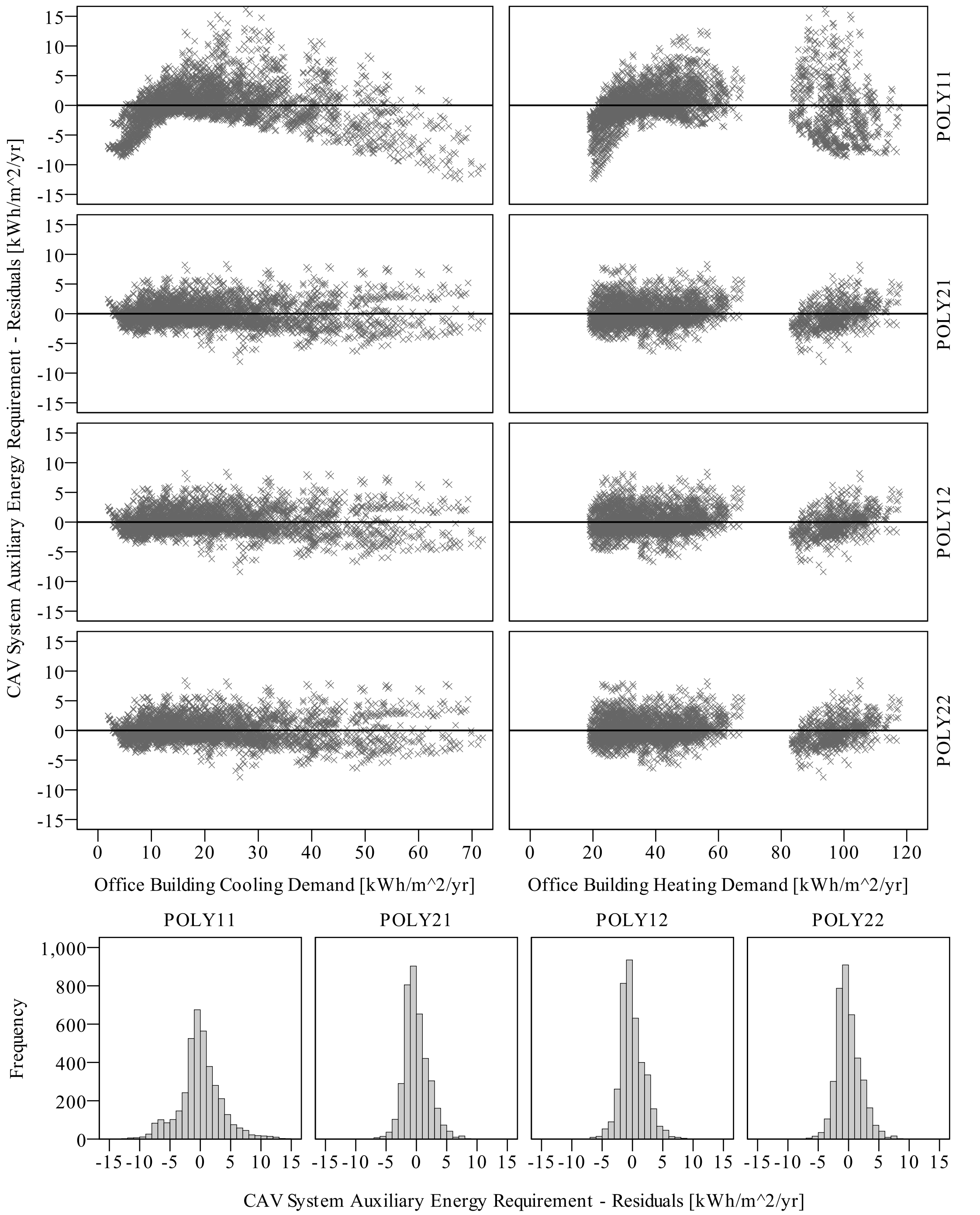


Figure C.7. Residuals scatter plots and histograms for two independent variables models of CAV system auxiliary energy consumption in office buildings

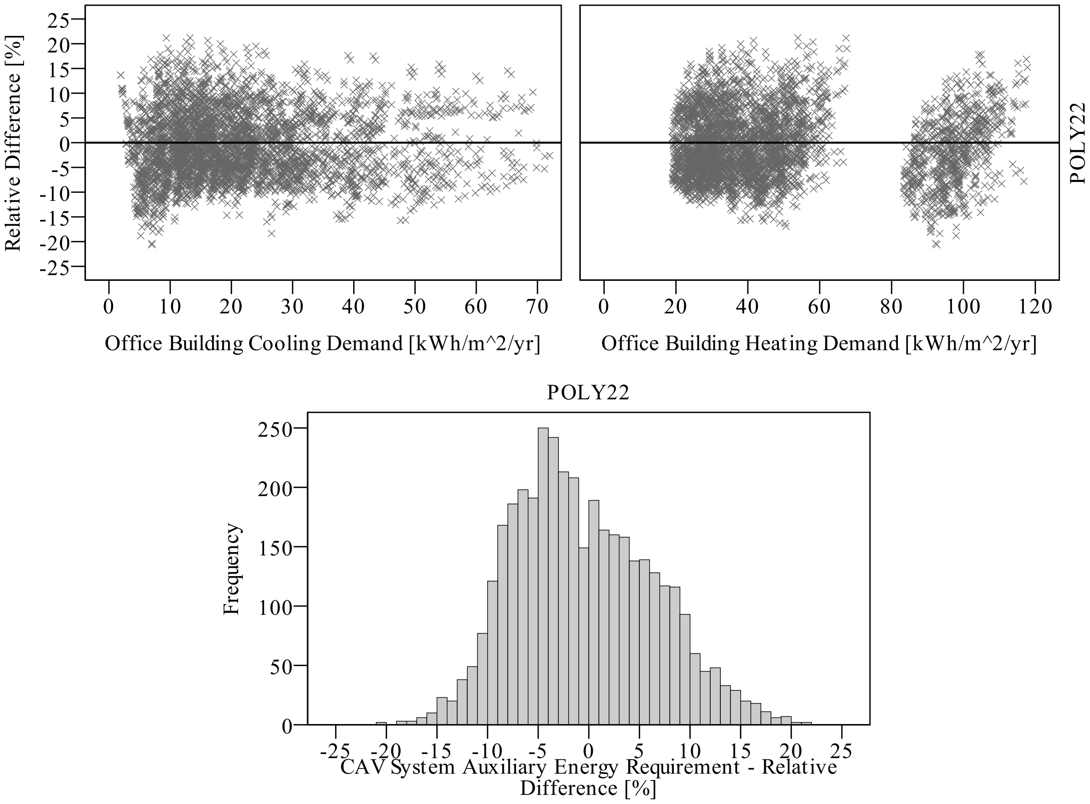


Figure C.8. Residuals scatter plots and histogram for POLY22 model of CAV system auxiliary energy consumption in office buildings

Table C.15. Comparison of relative differences between predicted and observed values for models of CAV system auxiliary energy consumption in office buildings

	POLY11	POLY21	POLY12	POLY22
Mean	-1.70	-0.55	-0.65	-0.54
Std. Dev.	12.684	6.917	7.022	6.911
Maximum	29.25	21.71	21.32	21.20
Minimum	-52.43	-20.23	-20.09	-20.63
Perc. 25	-7.71	-5.74	-5.86	-5.67
Perc. 75	6.58	4.37	4.43	4.38

C.3. Fan-coil system (FC)

- **Cooling energy requirements**

Table C.16. Regression model parameter values, standard errors and 95% confidence bounds for models of FC system cooling energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	1.411	6.697×10^{-4}	1.410 / 1.412
LIN2	a	1.122	2.670×10^{-2}	1.070 / 1.175
	b	1.373	1.054×10^{-3}	1.371 / 1.375
QUAD	a	6.861×10^{-1}	4.544×10^{-2}	5.970×10^{-1} / 7.752×10^{-1}
	b	1.414	3.642×10^{-3}	1.407 / 1.422
	c	-6.985×10^{-4}	5.942×10^{-5}	-8.150×10^{-4} / -5.820×10^{-4}
POW	a	2.920×10^{-1}	7.617×10^{-2}	1.426×10^{-1} / 4.413×10^{-1}
	b	1.54	1.487×10^{-2}	1.511 / 1.569
	c	9.732×10^{-1}	2.248×10^{-3}	9.688×10^{-1} / 9.776×10^{-1}
POLY11	a	2.793	4.121×10^{-2}	2.712 / 2.874
	b	1.349	9.812×10^{-4}	1.347 / 1.351
	c	-2.351×10^{-2}	4.971×10^{-4}	-2.449×10^{-2} / -2.254×10^{-2}
POLY21	a	7.601×10^{-1}	7.486×10^{-2}	6.134×10^{-1} / 9.069×10^{-1}
	b	1.501	4.962×10^{-3}	1.491 / 1.511
	c	5.861×10^{-3}	9.085×10^{-4}	4.079×10^{-3} / 7.642×10^{-3}
	d	-1.380×10^{-3}	5.710×10^{-5}	-1.491×10^{-3} / -1.268×10^{-3}
	e	-2.051×10^{-3}	5.517×10^{-5}	-2.159×10^{-3} / -1.943×10^{-3}
POLY12	a	4.901	9.495×10^{-2}	4.714 / 5.087
	b	1.352	1.932×10^{-3}	1.348 / 1.356
	c	-1.008×10^{-1}	3.131×10^{-3}	-1.070×10^{-1} / -9.469×10^{-2}
	e	-3.744×10^{-4}	4.865×10^{-5}	-4.698×10^{-4} / -2.790×10^{-4}
	f	6.268×10^{-4}	2.094×10^{-5}	5.858×10^{-4} / 6.679×10^{-4}
POLY22	a	3.376	1.315×10^{-1}	3.118 / 3.634
	b	1.435	5.426×10^{-3}	1.424 / 1.445
	c	-7.328×10^{-2}	3.473×10^{-3}	-8.009×10^{-2} / -6.647×10^{-2}
	d	-9.210×10^{-4}	5.685×10^{-5}	-1.032×10^{-3} / -8.096×10^{-4}
	e	-1.109×10^{-3}	6.534×10^{-5}	-1.237×10^{-3} / -9.805×10^{-4}
	f	5.069×10^{-4}	2.157×10^{-5}	4.646×10^{-4} / 5.492×10^{-4}

Table C.17. Comparison of models of FC system cooling energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	30.72	30.41	30.72	30.72	30.72	30.72	30.72	30.72	30.72
σ_y	18.324	18.805	18.303	18.304	18.304	18.310	18.314	18.315	18.315
RSS		4244	2907	2806	2803	1836	1350	1261	1180
R^2		0.9967	0.9977	0.9978	0.9978	0.9986	0.999	0.999	0.9991
RMSD		1.0513	0.8701	0.8548	0.8544	0.6915	0.5929	0.573	0.5543
e_{max}		3.32	3.28	3.11	3.14	2.81	2.12	2.08	1.93
e_{min}		-5.13	-3.86	-3.27	-3.32	-3.47	-3.22	-3.80	-3.31
$\overline{\lvert e \rvert}$		0.81	0.65	0.64	0.64	0.51	0.41	0.40	0.38
$\lvert e \rvert_{95\%}$		2.11	1.86	1.82	1.80	1.48	1.33	1.23	1.24

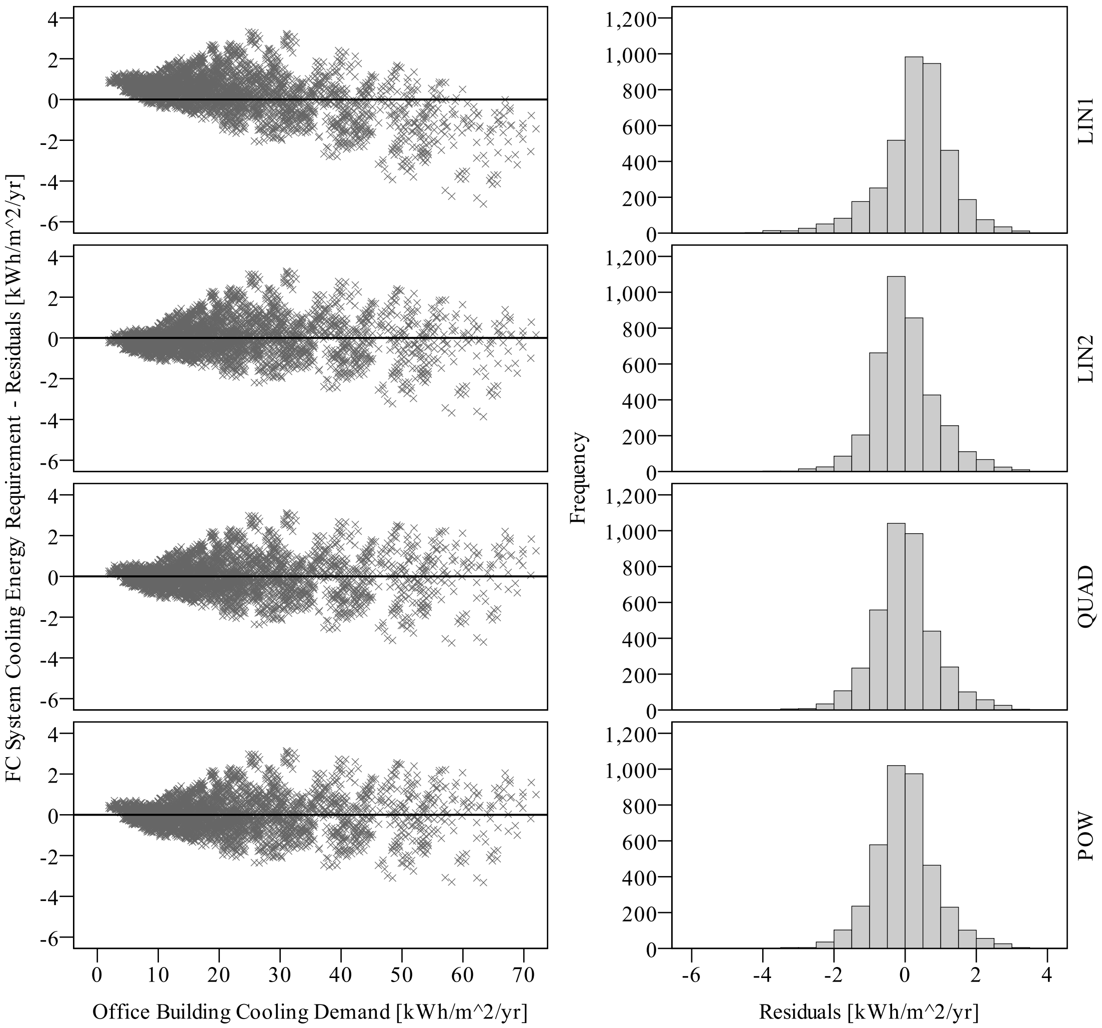


Figure C.9. Residuals scatter plots and histograms for single independent variable models of FC system cooling energy consumption in office buildings

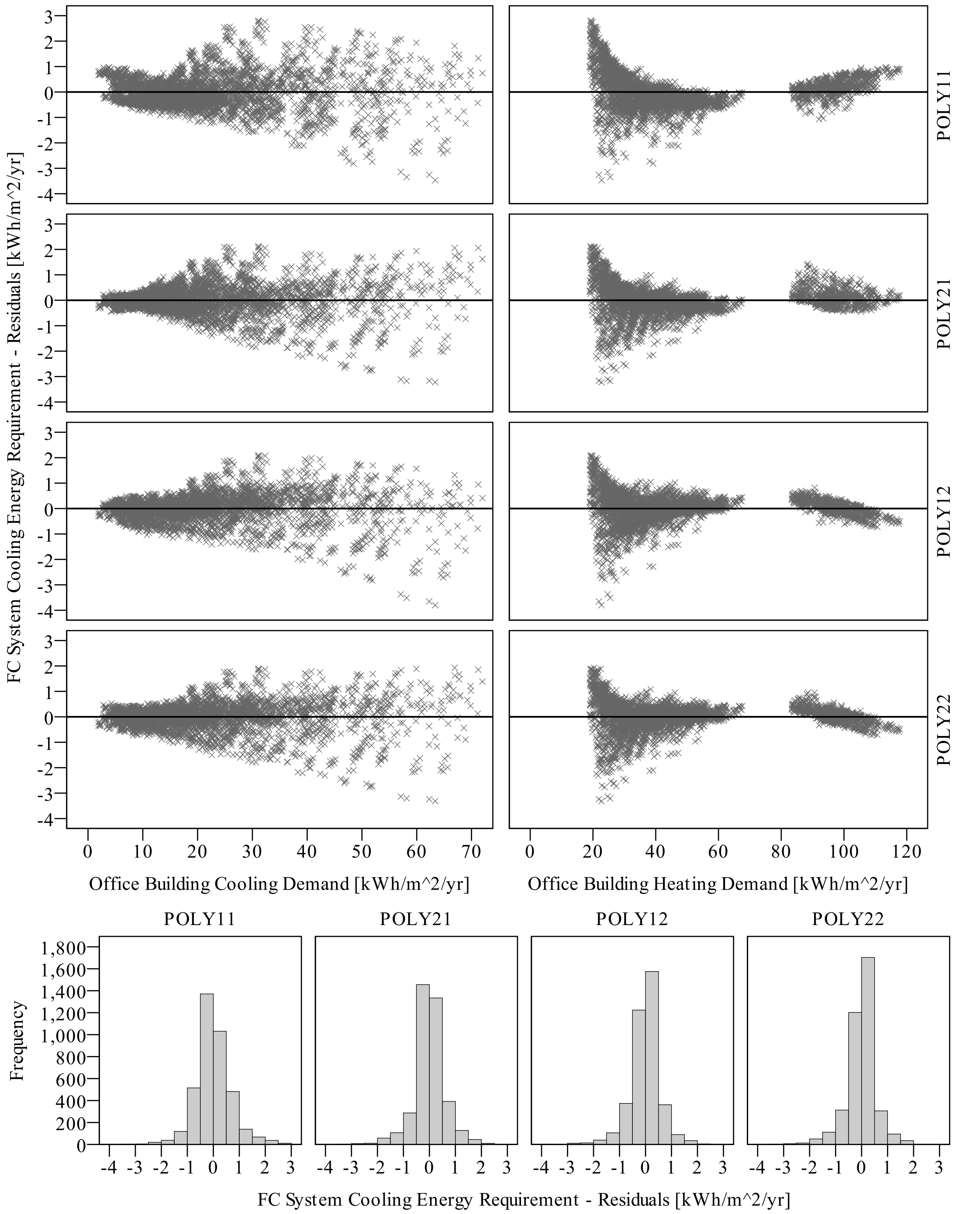


Figure C.10. Residuals scatter plots and histograms for two independent variables models of FC system cooling energy consumption in office buildings

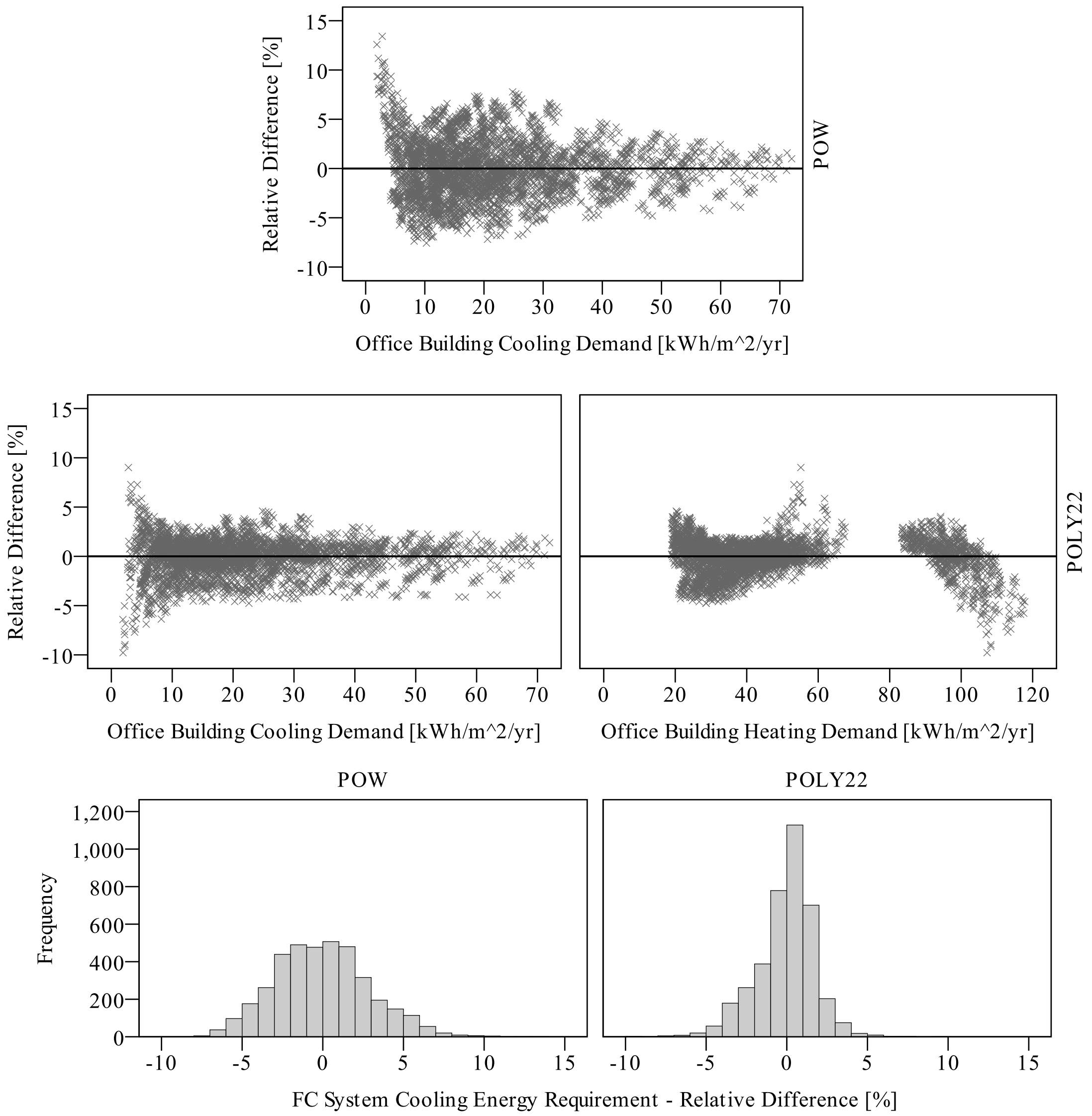


Figure C.11. Residuals scatter plots and histograms for POW and POLY22 models of FC system cooling energy consumption in office buildings

Table C.18. Comparison of relative differences between predicted and observed values for models of FC system cooling energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	2.31	-0.33	-0.07	-0.03	0.13	-0.08	-0.12	-0.04
Std. Dev.	3.968	2.959	2.842	2.925	2.842	1.762	1.852	1.793
Maximum	26.01	8.18	10.45	13.41	22.09	5.47	4.85	9.01
Minimum	-6.14	-9.13	-7.47	-7.57	-5.33	-9.02	-8.26	-9.78
Perc. 25	-0.45	-2.29	-2.11	-2.14	-1.77	-1.22	-1.04	-0.94
Perc. 75	4.63	1.57	1.77	1.80	1.39	1.04	1.05	1.07

- **Heating energy requirements**

Table C.19. Regression model parameter values, standard errors and 95% confidence bounds for models of FC system heating energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	6.027×10^{-1}	1.893×10^{-3}	5.989×10^{-1} / 6.064×10^{-1}
LIN2	a	-1.267×10^{1}	8.624×10^{-2}	-1.284×10^{1} / -1.250×10^{1}
	b	8.036×10^{-1}	1.553×10^{-3}	8.006×10^{-1} / 8.067×10^{-1}
QUAD	a	-4.815	2.041×10^{-1}	-5.215 / -4.415
	b	4.842×10^{-1}	7.875×10^{-3}	4.688×10^{-1} / 4.996×10^{-1}
	c	2.518×10^{-3}	6.125×10^{-5}	2.398×10^{-3} / 2.638×10^{-3}
POW	a	-1.774	2.116×10^{-1}	-2.189 / -1.359
	b	1.336×10^{-1}	6.095×10^{-3}	1.217×10^{-1} / 1.456×10^{-1}
	c	1.361	9.340×10^{-3}	1.343 / 1.380
POLY11	a	-1.425×10^{1}	1.476×10^{-1}	-1.454×10^{1} / -1.396×10^{1}
	b	4.577×10^{-2}	3.514×10^{-3}	3.888×10^{-2} / 5.266×10^{-2}
	c	8.157×10^{-1}	1.780×10^{-3}	8.122×10^{-1} / 8.192×10^{-1}
POLY21	a	-1.744×10^{1}	2.853×10^{-1}	-1.800×10^{1} / -1.688×10^{1}
	b	2.875×10^{-1}	1.891×10^{-2}	2.504×10^{-1} / 3.246×10^{-1}
	c	8.819×10^{-1}	3.463×10^{-3}	8.751×10^{-1} / 8.887×10^{-1}
	d	-1.238×10^{-3}	2.176×10^{-4}	-1.665×10^{-3} / -8.115×10^{-4}
	e	-5.116×10^{-3}	2.103×10^{-4}	-5.528×10^{-3} / -4.703×10^{-3}
POLY12	a	-7.548	3.448×10^{-1}	-8.223 / -6.872
	b	7.091×10^{-2}	7.016×10^{-3}	5.716×10^{-2} / 8.467×10^{-2}
	c	5.685×10^{-1}	1.137×10^{-2}	5.462×10^{-1} / 5.907×10^{-1}
	e	-1.691×10^{-3}	1.767×10^{-4}	-2.037×10^{-3} / -1.345×10^{-3}
	f	2.051×10^{-3}	7.604×10^{-5}	1.901×10^{-3} / 2.200×10^{-3}
POLY22	a	-6.391	4.929×10^{-1}	-7.357 / -5.424
	b	8.287×10^{-3}	2.034×10^{-2}	-3.158×10^{-2} / 4.816×10^{-2}
	c	5.476×10^{-1}	1.302×10^{-2}	5.220×10^{-1} / 5.731×10^{-1}
	d	6.989×10^{-4}	2.131×10^{-4}	2.812×10^{-4} / 1.117×10^{-3}
	e	-1.134×10^{-3}	2.449×10^{-4}	-1.614×10^{-3} / -6.536×10^{-4}
	f	2.142×10^{-3}	8.085×10^{-5}	1.983×10^{-3} / 2.300×10^{-3}

Table C.20. Comparison of models of FC system heating energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	26.63	29.47	26.63	26.63	26.63	26.63	26.63	26.63	26.63
σ_y	21.290	15.853	21.139	21.185	21.184	21.145	21.170	21.188	21.188
RSS		162891	24589	17068	17204	23548	19606	16620	16574
R^2		0.9064	0.9859	0.9902	0.9901	0.9865	0.9887	0.9904	0.9905
RMSD		6.513	2.5305	2.1083	2.1167	2.4763	2.2596	2.0804	2.0775
e_{max}		18.97	8.17	5.85	5.66	8.06	8.25	6.22	6.05
e_{min}		-8.41	-4.85	-5.11	-4.74	-4.52	-4.45	-4.88	-4.89
$\overline{\lvert e \rvert}$		5.88	2.08	1.73	1.73	2.03	1.83	1.71	1.70
$\lvert e \rvert_{95\%}$		10.87	4.64	4.11	4.08	4.30	4.16	4.01	4.01

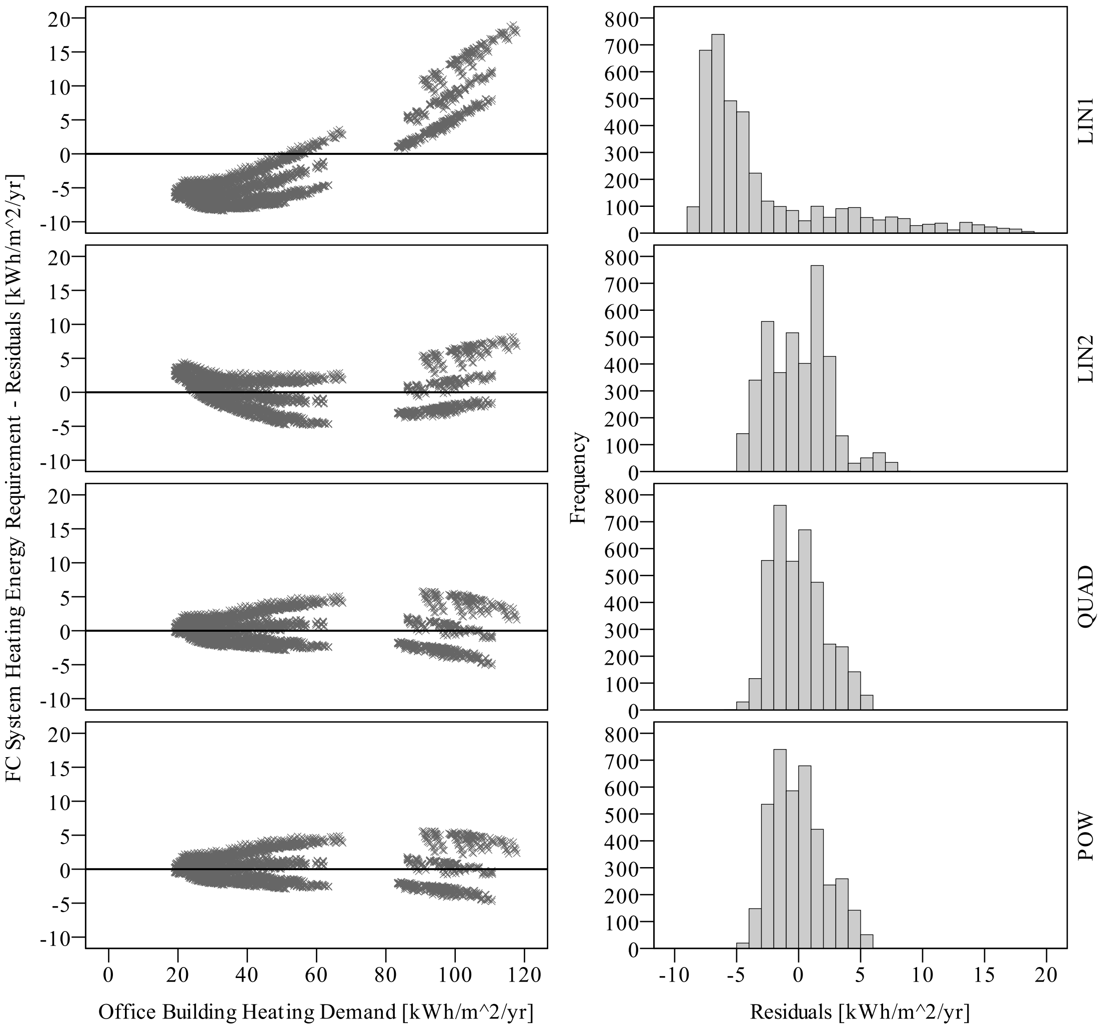


Figure C.12. Residuals scatter plots and histograms for single independent variable models of FC system heating energy consumption in office buildings

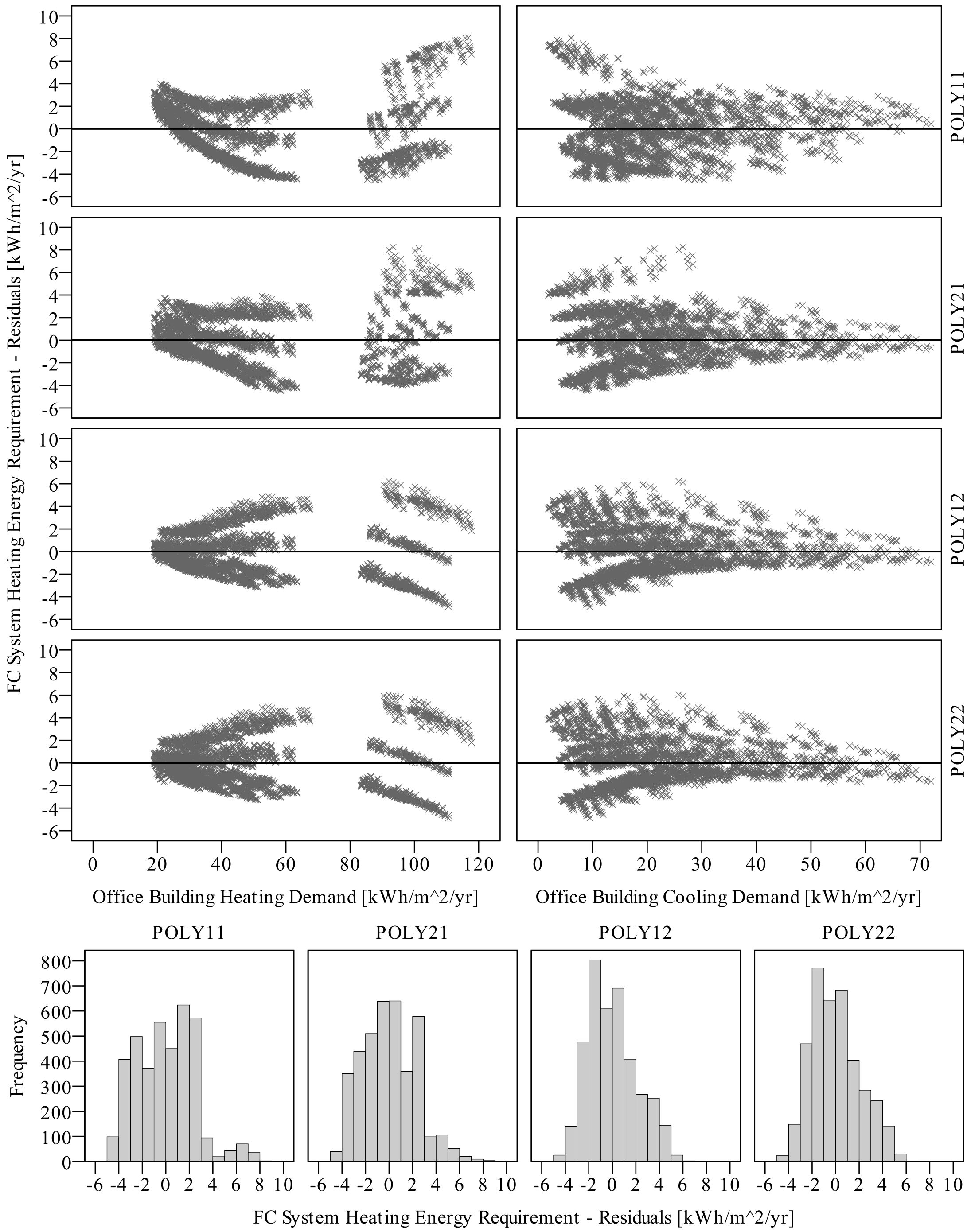


Figure C.13. Residuals scatter plots and histograms for two independent variables models of FC system heating energy consumption in office buildings

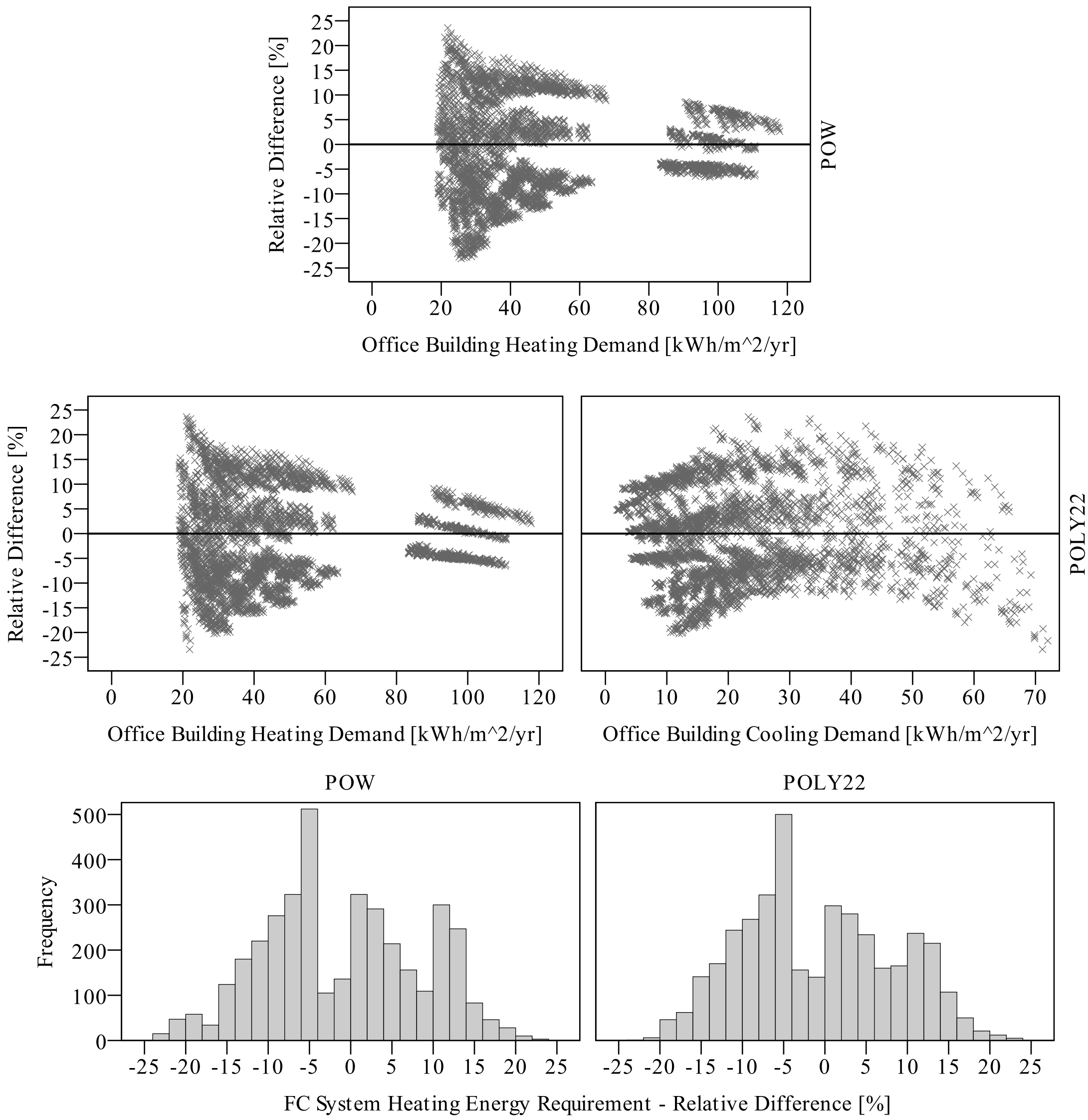


Figure C.14. Residuals scatter plots and histograms for POW and POLY22 models of FC system heating energy consumption in office buildings

Table C.21. Comparison of relative differences between predicted and observed values for models of FC system heating energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	-32.87	1.78	-0.86	-0.94	1.48	0.35	-0.87	-0.86
Std. Dev.	33.098	14.689	9.529	9.431	13.235	10.790	9.108	9.193
Maximum	21.23	56.16	25.25	23.57	55.44	43.19	22.99	23.60
Minimum	-124.00	-21.88	-23.05	-23.11	-18.88	-17.52	-18.54	-23.41
Perc. 25	-57.19	-7.06	-7.92	-7.95	-8.03	-8.74	-8.18	-7.90
Perc. 75	-4.59	8.34	6.46	6.25	8.62	7.68	5.95	6.16

- **Auxiliary energy requirements**

Table C.22. Regression model parameter values, standard errors and 95% confidence bounds for models of FC system auxiliary energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
POLY11	a	1.615	4.574×10^{-2}	1.525 / 1.705
	b	2.438×10^{-1}	1.089×10^{-3}	2.417×10^{-1} / 2.460×10^{-1}
	c	9.151×10^{-2}	5.518×10^{-4}	9.043×10^{-2} / 9.259×10^{-2}
POLY21	a	2.246	5.762×10^{-2}	2.134 / 2.359
	b	1.937×10^{-1}	3.819×10^{-3}	1.862×10^{-1} / 2.012×10^{-1}
	c	6.365×10^{-2}	6.993×10^{-4}	6.228×10^{-2} / 6.502×10^{-2}
	d	-4.402×10^{-4}	4.395×10^{-5}	-5.264×10^{-4} / -3.541×10^{-4}
	e	2.403×10^{-3}	4.246×10^{-5}	2.320×10^{-3} / 2.486×10^{-3}
POLY12	a	3.361	7.581×10^{-2}	3.212 / 3.509
	b	1.491×10^{-1}	1.543×10^{-3}	1.460×10^{-1} / 1.521×10^{-1}
	c	3.690×10^{-2}	2.499×10^{-3}	3.200×10^{-2} / 4.180×10^{-2}
	e	2.874×10^{-3}	3.885×10^{-5}	2.797×10^{-3} / 2.950×10^{-3}
	f	1.502×10^{-4}	1.672×10^{-5}	1.174×10^{-4} / 1.830×10^{-4}
POLY22	a	2.79	1.078×10^{-1}	2.578 / 3.001
	b	1.800×10^{-1}	4.446×10^{-3}	1.713×10^{-1} / 1.887×10^{-1}
	c	4.722×10^{-2}	2.846×10^{-3}	4.164×10^{-2} / 5.280×10^{-2}
	d	-3.450×10^{-4}	4.658×10^{-5}	-4.364×10^{-4} / -2.537×10^{-4}
	e	2.598×10^{-3}	5.354×10^{-5}	2.493×10^{-3} / 2.703×10^{-3}
	f	1.052×10^{-4}	1.768×10^{-5}	7.059×10^{-5} / 1.399×10^{-4}

Table C.23. Comparison of models of FC system auxiliary energy consumption in office buildings

	Observed	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	11.34	11.34	11.34	11.34	11.34
σ_y	2.966	2.865	2.931	2.931	2.931
RSS		2262	800	804	792
R^2		0.933	0.9763	0.9762	0.9765
RMSD		0.7675	0.4564	0.4576	0.4541
e_{max}		3.04	1.72	1.39	1.68
e_{min}		-3.31	-2.55	-2.77	-2.66
$\overline{\lvert e \rvert}$		0.56	0.35	0.35	0.34
$\lvert e \rvert_{95\%}$		1.70	0.88	0.86	0.87

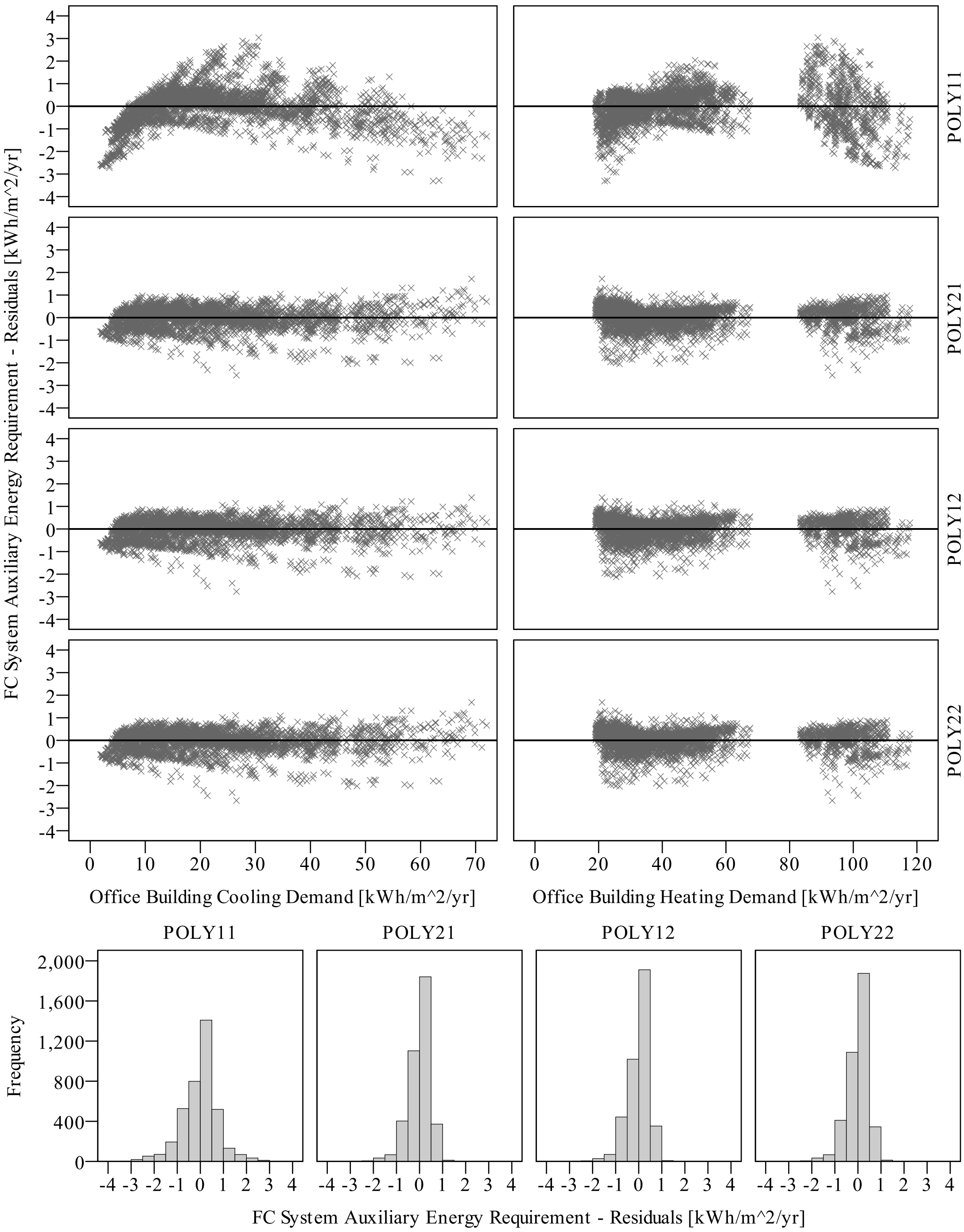
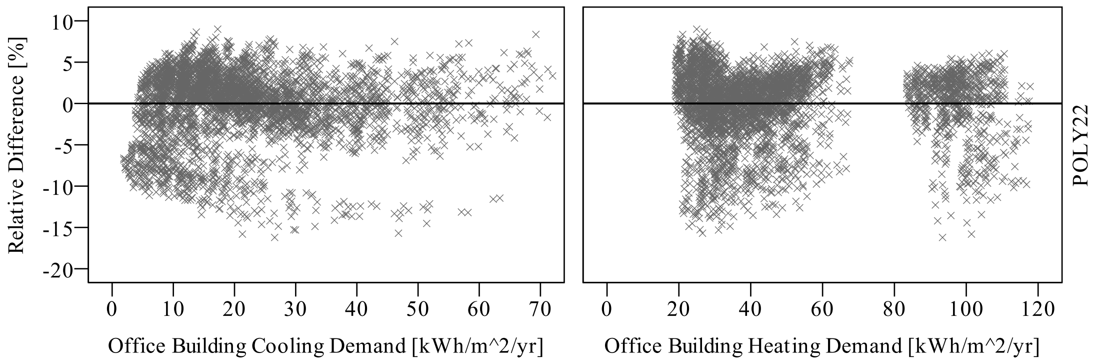


Figure C.15. Residuals scatter plots and histograms for two independent variables models of FC system auxiliary energy consumption in office buildings



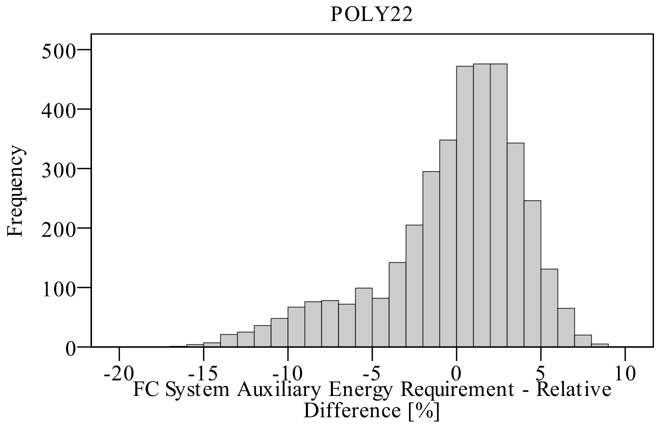


Figure C.16. Residuals scatter plots and histogram for POLY22 model of FC system auxiliary energy consumption in office buildings

Table C.24. Comparison of relative differences between predicted and observed values for models of FC system auxiliary energy consumption in office buildings

	POLY11	POLY21	POLY12	POLY22
Mean	-0.45	-0.19	-0.23	-0.20
Std. Dev.	6.518	4.258	4.334	4.244
Maximum	15.01	9.82	8.67	8.99
Minimum	-28.26	-15.78	-16.85	-16.21
Perc. 25	-3.94	-2.09	-2.00	-2.01
Perc. 75	3.93	2.64	2.65	2.67

C.4. Chilled ceiling system – embedded pipes (EMB)

- **Cooling energy requirements**

Table C.25. Regression model parameter values, standard errors and 95% confidence bounds for models of EMB system cooling energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	1.513	2.526×10^{-3}	1.508 / 1.518
LIN2	a	3.814	1.050×10^{-1}	3.609 / 4.020
	b	1.385	4.143×10^{-3}	1.377 / 1.393
QUAD	a	-1.298	1.511×10^{-1}	-1.594 / -1.002
	b	1.867	1.211×10^{-2}	1.843 / 1.890
	c	-8.189×10^{-3}	1.976×10^{-4}	-8.576×10^{-3} / -7.801×10^{-3}
POW	a	-8.974	4.304×10^{-1}	-9.817 / -8.130
	b	5.073	1.595×10^{-1}	4.761 / 5.386
	c	7.055×10^{-1}	6.870×10^{-3}	6.920×10^{-1} / 7.190×10^{-1}
POLY11	a	5.134×10^{-1}	1.941×10^{-1}	1.329×10^{-1} / 8.940×10^{-1}
	b	1.433	4.622×10^{-3}	1.424 / 1.442
	c	4.646×10^{-2}	2.342×10^{-3}	4.187×10^{-2} / 5.105×10^{-2}
POLY21	a	-1.986	2.478×10^{-1}	-2.472 / -1.500
	b	1.604	1.643×10^{-2}	1.572 / 1.636
	c	-1.370×10^{-2}	3.008×10^{-3}	-1.960×10^{-2} / -7.802×10^{-3}
	d	-6.162×10^{-3}	1.890×10^{-4}	-6.533×10^{-3} / -5.792×10^{-3}
	e	6.554×10^{-3}	1.826×10^{-4}	6.196×10^{-3} / 6.912×10^{-3}
POLY12	a	1.013×10^{1}	3.557×10^{-1}	9.432 / 1.083×10^{1}
	b	1.026	7.239×10^{-3}	1.012 / 1.040
	c	-2.655×10^{-1}	1.173×10^{-2}	-2.885×10^{-1} / -2.425×10^{-1}
	e	1.206×10^{-2}	1.823×10^{-4}	1.170×10^{-2} / 1.242×10^{-2}
	f	1.264×10^{-3}	7.845×10^{-5}	1.110×10^{-3} / 1.418×10^{-3}
POLY22	a	7.126×10^{-1}	4.628×10^{-1}	-1.948×10^{-1} / 1.620
	b	1.536	1.909×10^{-2}	1.499 / 1.573
	c	-9.533×10^{-2}	1.222×10^{-2}	-1.193×10^{-1} / -7.136×10^{-2}
	d	-5.689×10^{-3}	2.001×10^{-4}	-6.082×10^{-3} / -5.297×10^{-3}
	e	7.526×10^{-3}	2.300×10^{-4}	7.076×10^{-3} / 7.977×10^{-3}
	f	5.228×10^{-4}	7.592×10^{-5}	3.740×10^{-4} / 6.717×10^{-4}

Table C.26. Comparison of models of EMB system cooling energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	33.67	32.61	33.67	33.67	33.67	33.67	33.67	33.67	33.67
σ_y	18.777	20.169	18.463	18.561	18.561	18.492	18.674	18.654	18.675
RSS		60373	44917	31031	30909	40737	14793	17695	14612
R^2		0.9554	0.9668	0.9771	0.9772	0.9699	0.9891	0.9869	0.9892
RMSD		3.9651	3.4201	2.8427	2.8371	3.2571	1.9627	2.1466	1.9507
e_{max}		15.20	14.90	12.98	13.23	12.87	10.12	8.06	9.97
e_{min}		-19.16	-14.64	-9.07	-8.43	-15.51	-9.37	-10.18	-9.32
$\overline{\lvert e\rvert}$		2.74	2.56	2.09	2.13	2.42	1.41	1.60	1.41
$\lvert e\rvert_{95\%}$		8.93	7.47	6.11	5.97	6.73	4.18	4.62	4.12

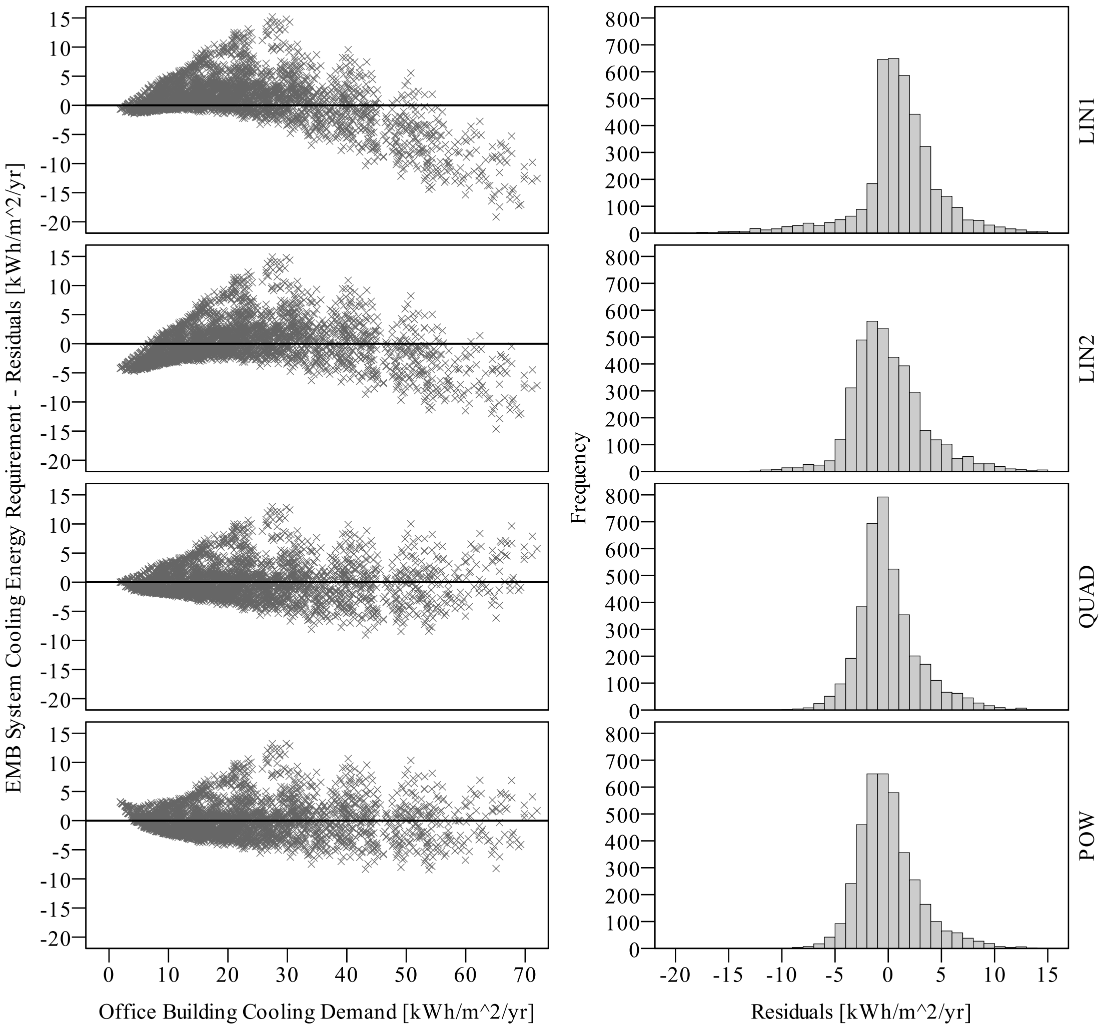


Figure C.17. Residuals scatter plots and histograms for single independent variable models of EMB system cooling energy consumption in office buildings

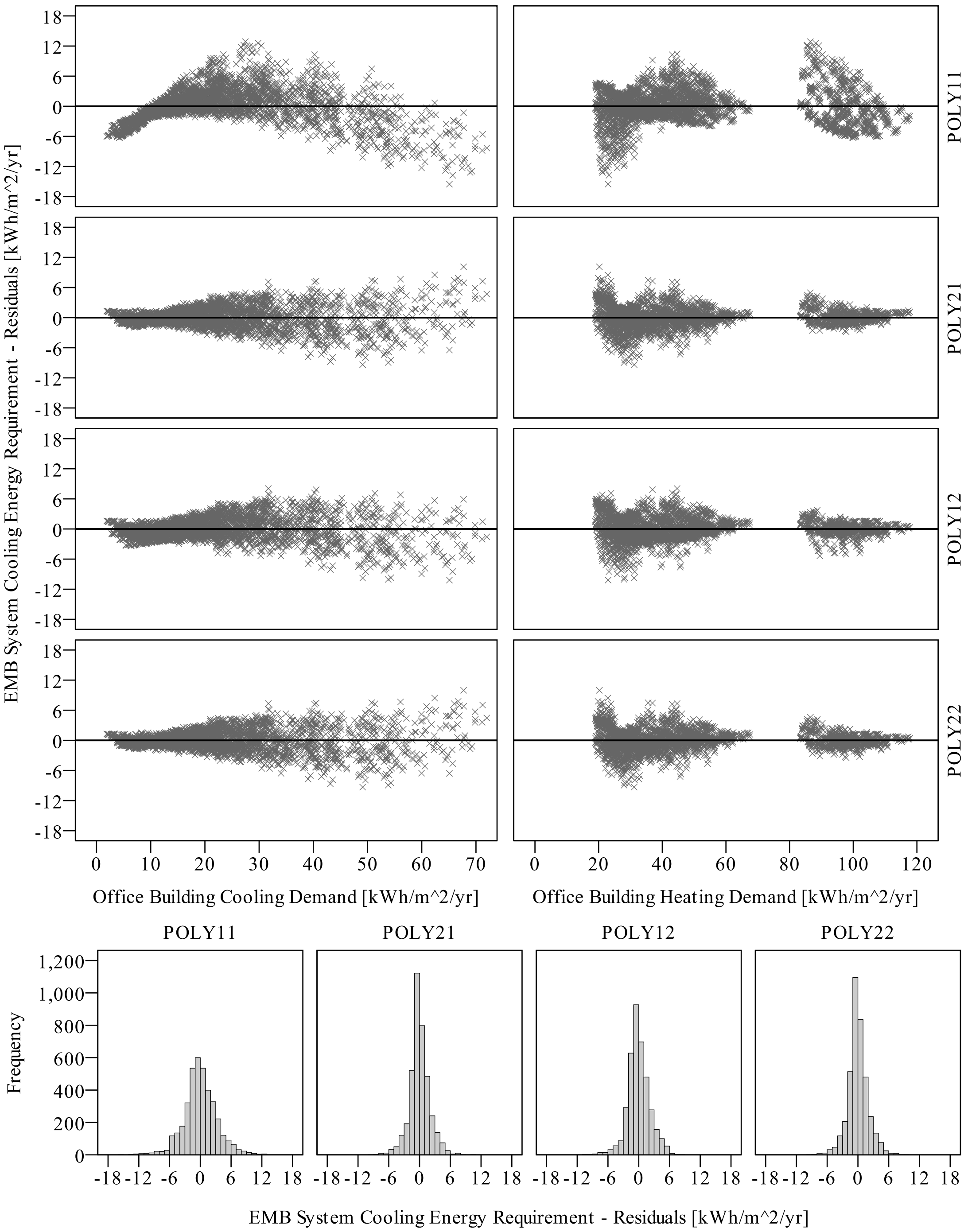


Figure C.18. Residuals scatter plots and histograms for two independent variables models of EMB system cooling energy consumption in office buildings

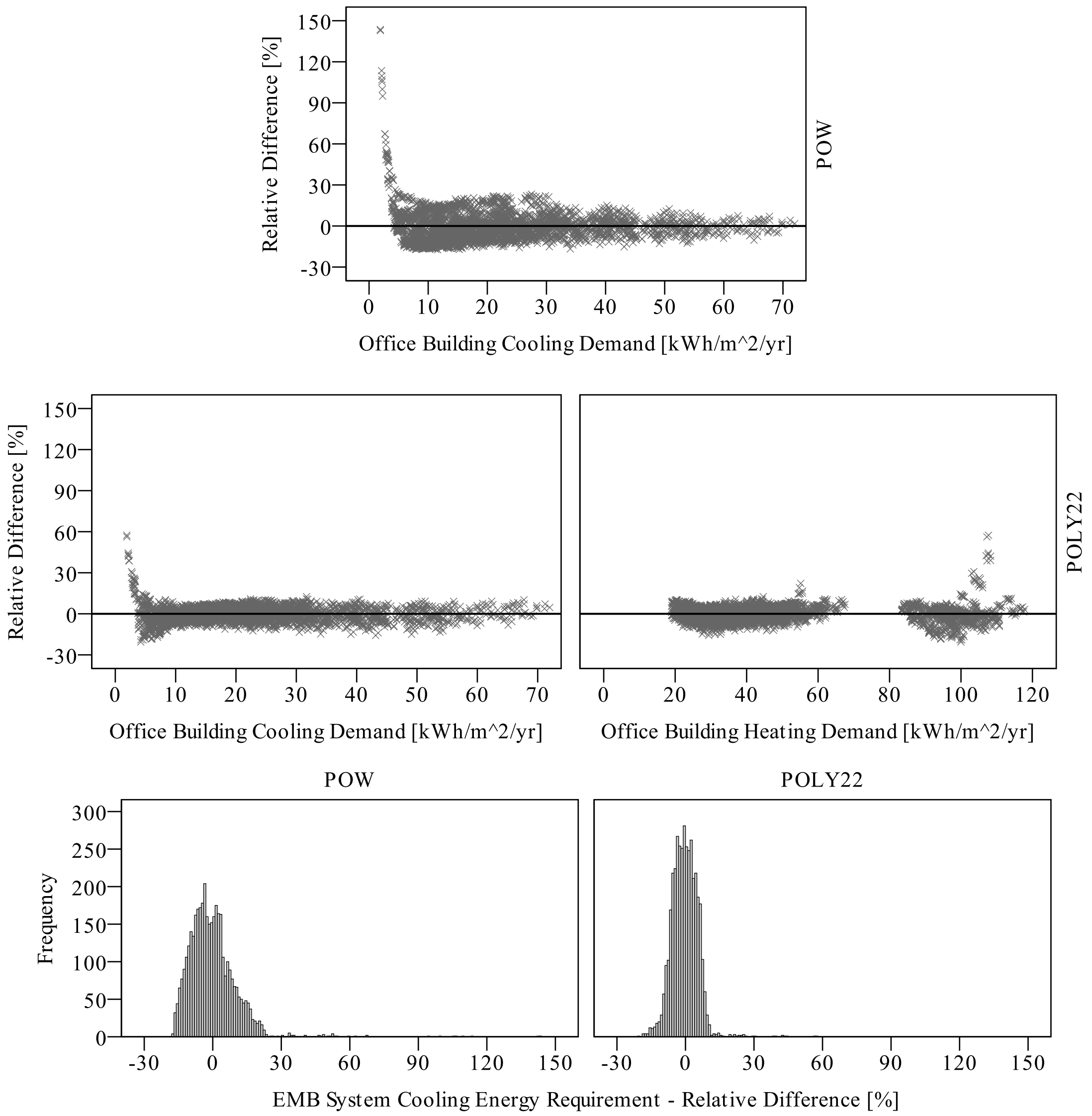


Figure C.19. Residuals scatter plots and histograms for POW and POLY22 models of EMB system cooling energy consumption in office buildings

Table C.27. Comparison of relative differences between predicted and observed values for models of EMB system cooling energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	4.16	-4.53	-1.08	-0.22	-5.15	-0.38	-0.96	-0.32
Std. Dev.	9.655	18.287	8.613	10.842	21.685	5.889	7.899	5.790
Maximum	28.23	26.25	22.86	143.55	22.67	59.30	72.20	57.42
Minimum	-35.93	-192.06	-30.15	-17.31	-272.13	-22.73	-46.84	-20.57
Perc. 25	-1.99	-8.36	-7.21	-7.06	-7.15	-4.00	-4.88	-4.11
Perc. 75	9.79	4.94	3.93	4.47	5.19	3.40	3.94	3.44

- **Heating energy requirements**

Table C.28. Regression model parameter values, standard errors and 95% confidence bounds for models of EMB system heating energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	5.815×10^{-1}	9.389×10^{-4}	5.796×10^{-1} / 5.833×10^{-1}
LIN2	a	-4.667	8.028×10^{-2}	-4.825 / -4.510
	b	6.555×10^{-1}	1.446×10^{-3}	6.527×10^{-1} / 6.583×10^{-1}
QUAD	a	-1.827	2.227×10^{-1}	-2.264 / -1.390
	b	5.400×10^{-1}	8.594×10^{-3}	5.231×10^{-1} / 5.568×10^{-1}
	c	9.106×10^{-4}	6.683×10^{-5}	7.795×10^{-4} / 1.042×10^{-3}
POW	a	-4.637×10^{-1}	3.120×10^{-1}	-1.075 / 1.479×10^{-1}
	b	3.218×10^{-1}	1.882×10^{-2}	2.849×10^{-1} / 3.587×10^{-1}
	c	1.141	1.172×10^{-2}	1.118 / 1.164
POLY11	a	-6.831	1.338×10^{-1}	-7.093 / -6.569
	b	6.280×10^{-2}	3.185×10^{-3}	5.656×10^{-2} / 6.905×10^{-2}
	c	6.721×10^{-1}	1.614×10^{-3}	6.689×10^{-1} / 6.752×10^{-1}
POLY21	a	-6.872	2.833×10^{-1}	-7.428 / -6.317
	b	6.617×10^{-2}	1.878×10^{-2}	2.935×10^{-2} / 1.030×10^{-1}
	c	6.745×10^{-1}	3.439×10^{-3}	6.677×10^{-1} / 6.812×10^{-1}
	d	5.590×10^{-5}	2.161×10^{-4}	-3.678×10^{-4} / 4.796×10^{-4}
	e	-2.119×10^{-4}	2.088×10^{-4}	-6.213×10^{-4} / 1.974×10^{-4}
POLY12	a	-3.784	3.677×10^{-1}	-4.505 / -3.063
	b	2.835×10^{-2}	7.482×10^{-3}	1.368×10^{-2} / 4.302×10^{-2}
	c	5.640×10^{-1}	1.212×10^{-2}	5.403×10^{-1} / 5.878×10^{-1}
	e	7.319×10^{-4}	1.884×10^{-4}	3.625×10^{-4} / 1.101×10^{-3}
	f	7.587×10^{-4}	8.109×10^{-5}	5.997×10^{-4} / 9.177×10^{-4}
POLY22	a	-2.391	5.254×10^{-1}	-3.421 / -1.361
	b	-4.703×10^{-2}	2.168×10^{-2}	-8.953×10^{-2} / -4.527×10^{-3}
	c	5.389×10^{-1}	1.388×10^{-2}	5.117×10^{-1} / 5.661×10^{-1}
	d	8.413×10^{-4}	2.271×10^{-4}	3.960×10^{-4} / 1.287×10^{-3}
	e	1.402×10^{-3}	2.611×10^{-4}	8.906×10^{-4} / 1.914×10^{-3}
	f	8.683×10^{-4}	8.619×10^{-5}	6.993×10^{-4} / 1.037×10^{-3}

Table C.29. Comparison of models of EMB system heating energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	27.39	28.44	27.39	27.39	27.39	27.39	27.39	27.39	27.39
σ_y	17.403	15.295	17.242	17.250	17.248	17.257	17.257	17.261	17.261
RSS		40074	21307	20324	20523	19346	19334	18903	18835
R^2		0.9655	0.9817	0.9825	0.9823	0.9834	0.9834	0.9837	0.9838
RMSD		3.2305	2.3556	2.3006	2.3118	2.2446	2.2439	2.2187	2.2147
e_{max}		12.85	8.80	7.03	7.45	8.56	8.47	7.38	7.36
e_{min}		-4.94	-6.04	-5.67	-5.81	-5.91	-5.91	-5.52	-5.47
$\overline{\lvert e \rvert}$		2.66	1.81	1.76	1.76	1.71	1.70	1.68	1.68
$\lvert e \rvert_{95\%}$		5.43	5.04	5.17	5.20	4.89	4.92	4.94	4.84

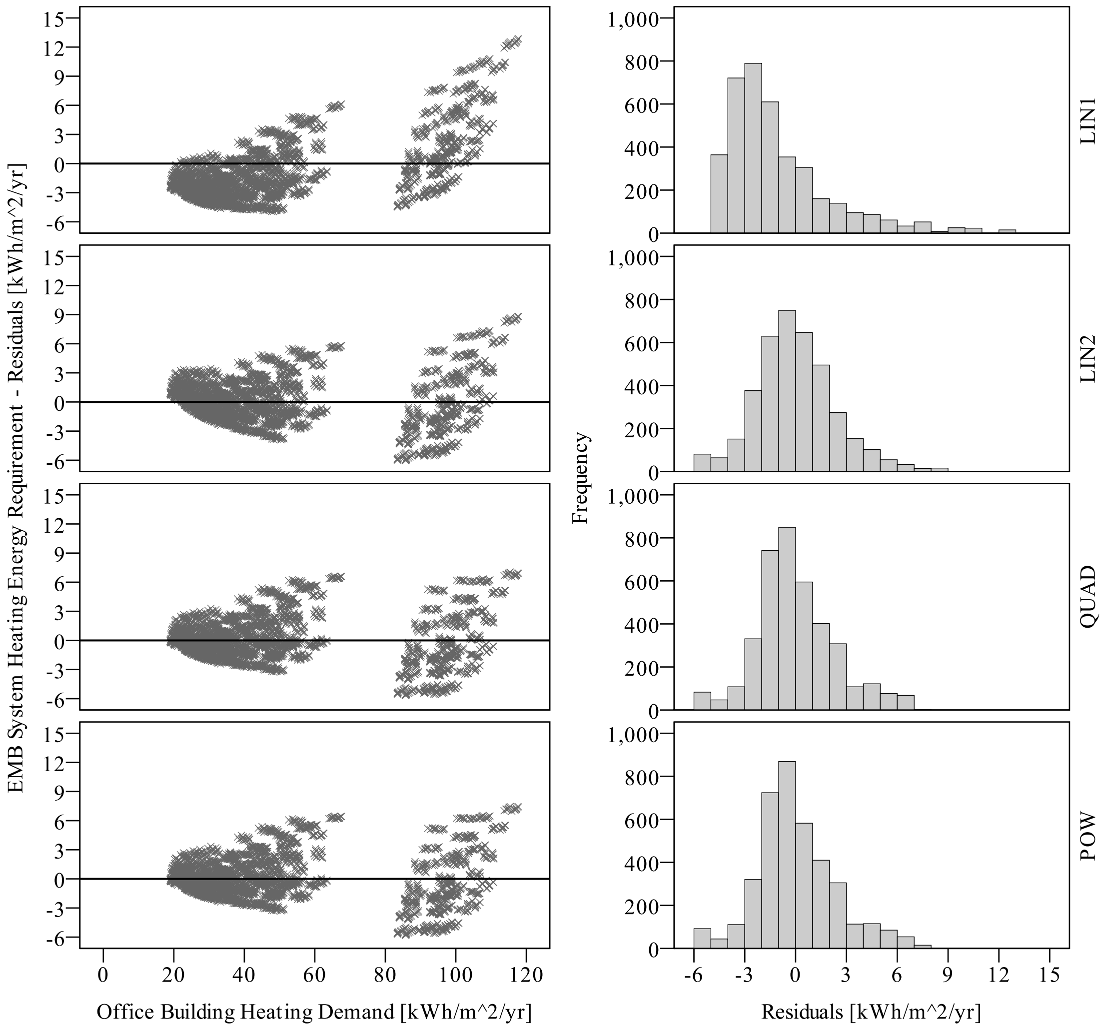


Figure C.20. Residuals scatter plots and histograms for single independent variable models of EMB system heating energy consumption in office buildings

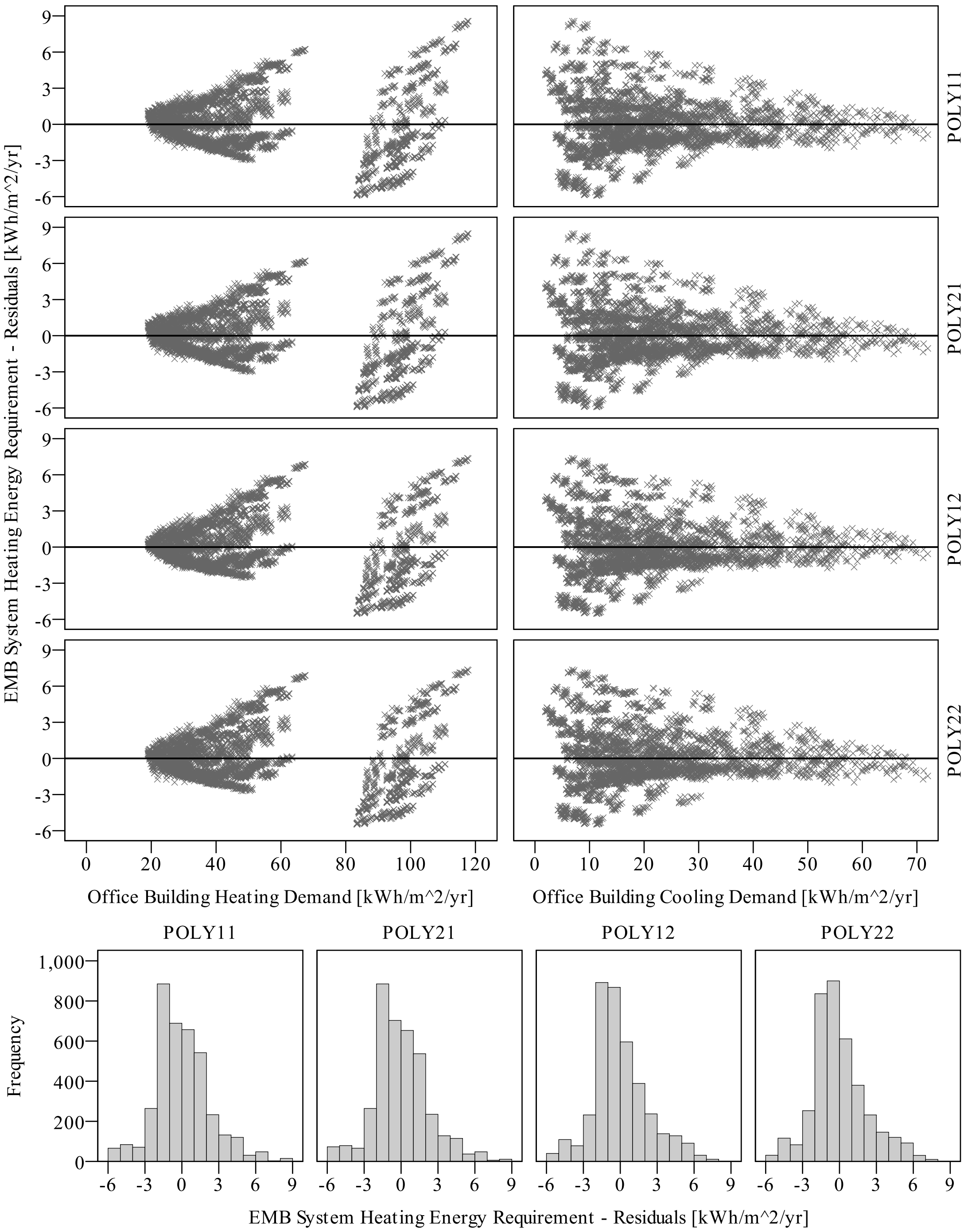


Figure C.21. Residuals scatter plots and histograms for two independent variables models of EMB system heating energy consumption in office buildings

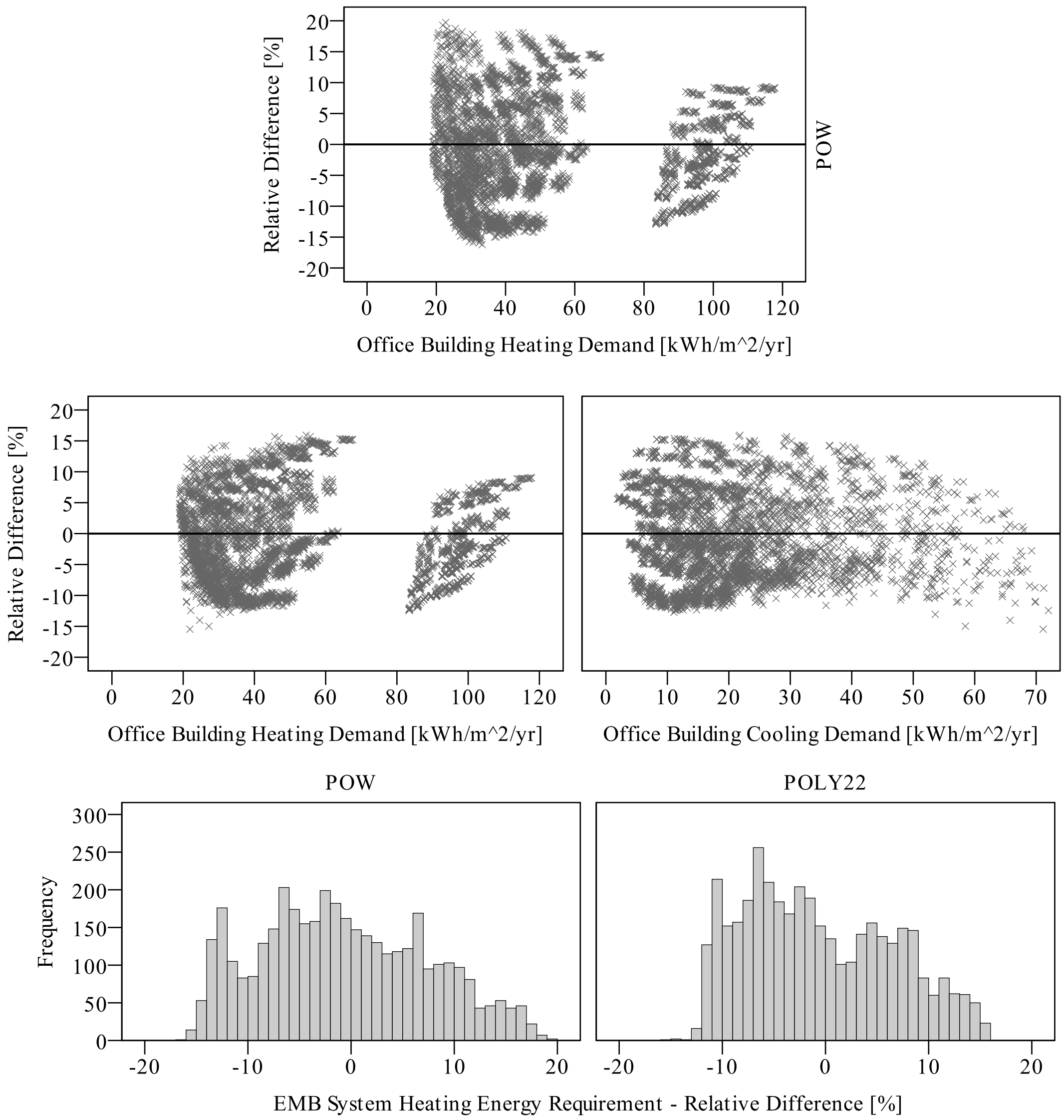


Figure C.22. Residuals scatter plots and histograms for POW and POLY22 models of EMB system heating energy consumption in office buildings

Table C.30. Comparison of relative differences between predicted and observed values for models of EMB system heating energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	-9.70	-0.22	-0.76	-0.72	-0.34	-0.37	-0.67	-0.67
Std. Dev.	12.490	8.579	8.133	8.096	7.242	7.229	7.160	7.172
Maximum	15.80	25.36	19.69	19.73	14.35	14.52	15.63	15.91
Minimum	-34.68	-16.65	-16.48	-16.23	-13.89	-14.45	-12.85	-15.47
Perc. 25	-19.94	-6.56	-6.87	-6.84	-6.75	-6.69	-6.82	-6.53
Perc. 75	0.39	5.75	5.44	5.46	6.45	6.29	5.56	5.17

- **Auxiliary energy requirements**

Table C.31. Regression model parameter values, standard errors and 95% confidence bounds for models of EMB system auxiliary energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
POLY11	a	4.434	3.394×10^{-2}	4.367 / 4.500
	b	1.121×10^{-1}	8.083×10^{-4}	1.106×10^{-1} / 1.137×10^{-1}
	c	1.791×10^{-2}	4.095×10^{-4}	1.711×10^{-2} / 1.871×10^{-2}
POLY21	a	3.68	6.445×10^{-2}	3.554 / 3.806
	b	1.667×10^{-1}	4.272×10^{-3}	1.584×10^{-1} / 1.751×10^{-1}
	c	1.849×10^{-2}	7.821×10^{-4}	1.696×10^{-2} / 2.003×10^{-2}
	d	-9.902×10^{-4}	4.916×10^{-5}	-1.087×10^{-3} / -8.939×10^{-4}
	e	2.116×10^{-4}	4.749×10^{-5}	1.184×10^{-4} / 3.047×10^{-4}
POLY12	a	5.334	8.842×10^{-2}	5.161 / 5.508
	b	7.782×10^{-2}	1.799×10^{-3}	7.429×10^{-2} / 8.135×10^{-2}
	c	-1.166×10^{-2}	2.915×10^{-3}	-1.737×10^{-2} / -5.944×10^{-3}
	e	1.006×10^{-3}	4.530×10^{-5}	9.168×10^{-4} / 1.094×10^{-3}
	f	1.327×10^{-4}	1.950×10^{-5}	9.442×10^{-5} / 1.709×10^{-4}
POLY22	a	3.701	1.211×10^{-1}	3.464 / 3.939
	b	1.662×10^{-1}	4.996×10^{-3}	1.564×10^{-1} / 1.760×10^{-1}
	c	1.784×10^{-2}	3.198×10^{-3}	1.157×10^{-2} / 2.411×10^{-2}
	d	-9.865×10^{-4}	5.234×10^{-5}	-1.089×10^{-3} / -8.838×10^{-4}
	e	2.193×10^{-4}	6.017×10^{-5}	1.013×10^{-4} / 3.373×10^{-4}
	f	4.163×10^{-6}	1.986×10^{-5}	-3.478×10^{-5} / 4.311×10^{-5}

Table C.32. Comparison of models of EMB system auxiliary energy consumption in office buildings

	Observed	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	7.73	7.73	7.73	7.73	7.73
σ_y	1.431	1.312	1.336	1.327	1.336
RSS		1246	1000	1093	1000
R^2		0.8414	0.8727	0.8609	0.8727
RMSD		0.5696	0.5103	0.5335	0.5103
e_{max}		1.16	1.35	0.75	1.35
e_{min}		-2.17	-1.70	-1.72	-1.70
$\overline{\lvert e \rvert}$		0.48	0.42	0.45	0.42
$\lvert e \rvert_{95\%}$		1.03	0.94	0.99	0.94

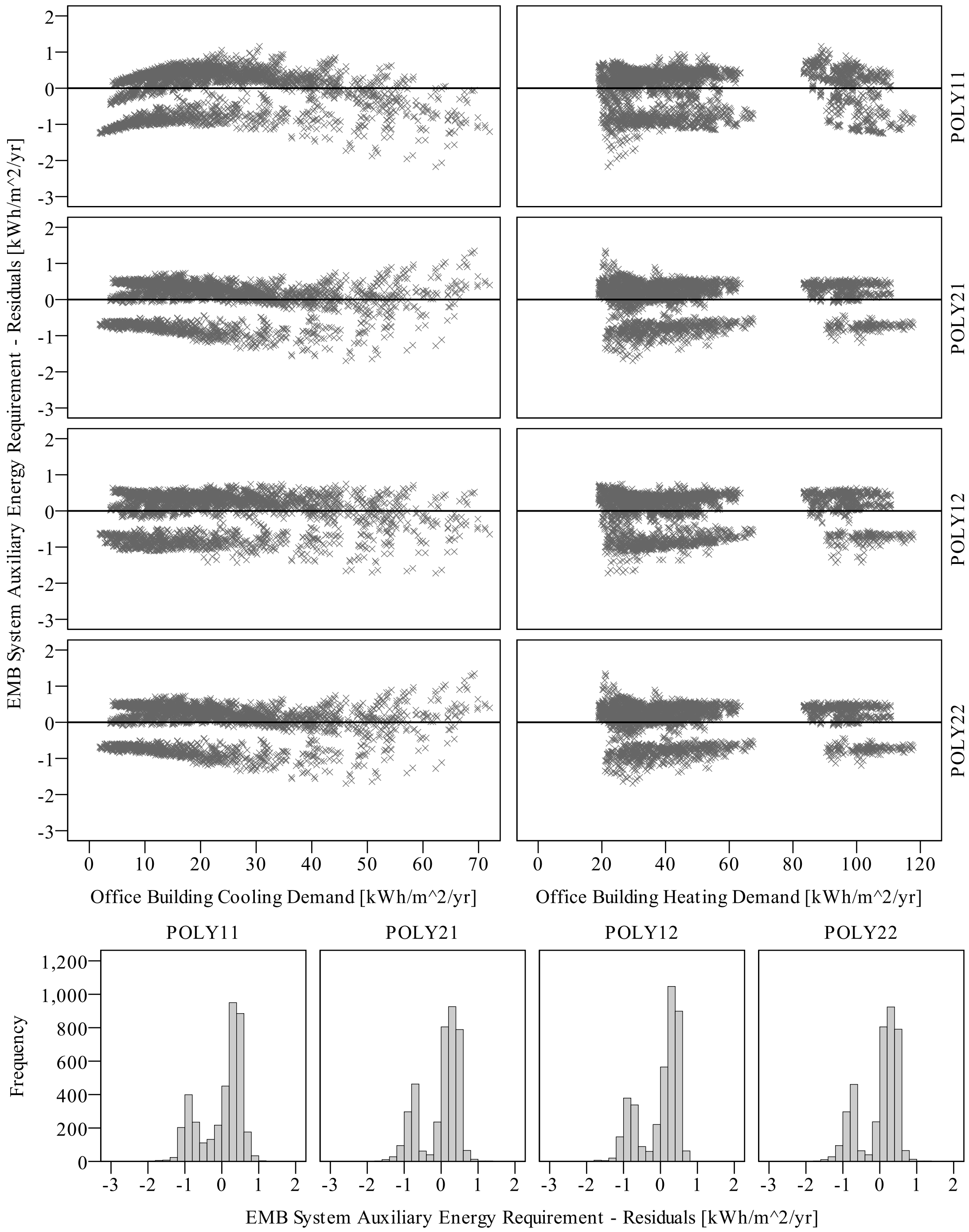
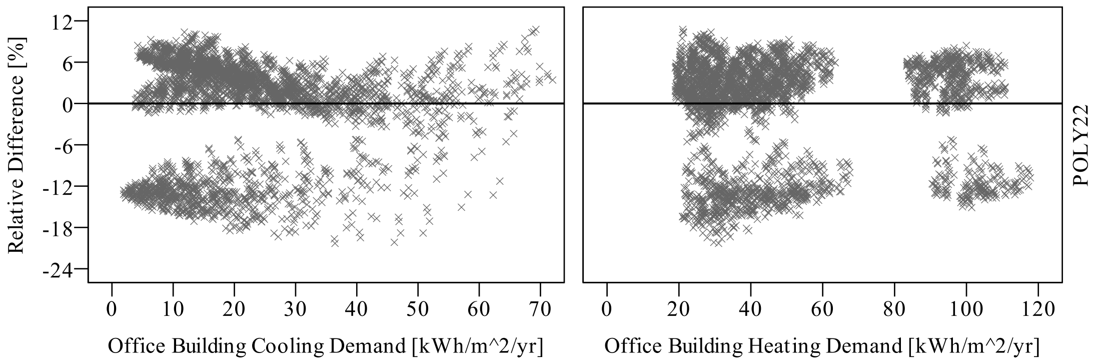


Figure C.23. Residuals scatter plots and histograms for two independent variables models of EMB system auxiliary energy consumption in office buildings



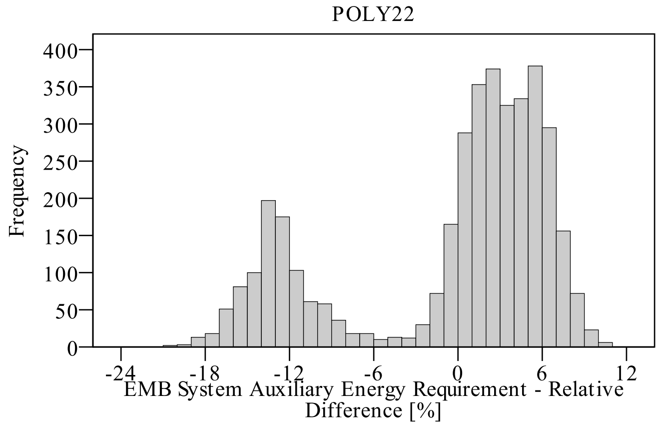


Figure C.24. Residuals scatter plots and histogram for POLY22 model of EMB system auxiliary energy consumption in office buildings

Table C.33. Comparison of relative differences between predicted and observed values for models of EMB system auxiliary energy consumption in office buildings

	POLY11	POLY21	POLY12	POLY22
Mean	-0.75	-0.55	-0.65	-0.55
Std. Dev.	8.467	7.459	7.969	7.460
Maximum	10.92	10.77	9.24	10.76
Minimum	-25.20	-20.32	-23.26	-20.31
Perc. 25	-5.74	-3.61	-5.35	-3.61
Perc. 75	5.42	4.92	5.12	4.92

C.5. Chilled ceiling system – aluminium panels (ALU)

- **Cooling energy requirements**

Table C.34. Regression model parameter values, standard errors and 95% confidence bounds for models of ALU system cooling energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	1.279	2.058×10^{-3}	1.274 / 1.283
LIN2	a	3.589	8.046×10^{-2}	3.431 / 3.746
	b	1.158	3.176×10^{-3}	1.152 / 1.164
QUAD	a	7.355×10^{-1}	1.274×10^{-1}	4.856×10^{-1} / 9.853×10^{-1}
	b	1.427	1.021×10^{-2}	1.407 / 1.447
	c	-4.570×10^{-3}	1.667×10^{-4}	-4.897×10^{-3} / -4.243×10^{-3}
POW	a	-3.53	3.002×10^{-1}	-4.118 / -2.941
	b	2.997	9.350×10^{-2}	2.814 / 3.180
	c	7.820×10^{-1}	6.980×10^{-3}	7.683×10^{-1} / 7.957×10^{-1}
POLY11	a	5.678×10^{-1}	1.455×10^{-1}	2.825×10^{-1} / 8.531×10^{-1}
	b	1.202	3.465×10^{-3}	1.195 / 1.209
	c	4.252×10^{-2}	1.755×10^{-3}	3.908×10^{-2} / 4.596×10^{-2}
POLY21	a	2.568	1.879×10^{-1}	2.200 / 2.937
	b	1.043	1.246×10^{-2}	1.019 / 1.067
	c	-4.505×10^{-2}	2.281×10^{-3}	-4.952×10^{-2} / -4.057×10^{-2}
	d	-1.364×10^{-3}	1.434×10^{-4}	-1.645×10^{-3} / -1.082×10^{-3}
	e	7.548×10^{-3}	1.385×10^{-4}	7.277×10^{-3} / 7.820×10^{-3}
POLY12	a	8.4	2.377×10^{-1}	7.934 / 8.866
	b	8.727×10^{-1}	4.837×10^{-3}	8.633×10^{-1} / 8.822×10^{-1}
	c	-2.118×10^{-1}	7.837×10^{-3}	-2.271×10^{-1} / -1.964×10^{-1}
	e	9.746×10^{-3}	1.218×10^{-4}	9.507×10^{-3} / 9.985×10^{-3}
	f	1.038×10^{-3}	5.242×10^{-5}	9.356×10^{-4} / 1.141×10^{-3}
POLY22	a	7.604	3.398×10^{-1}	6.937 / 8.270
	b	9.158×10^{-1}	1.402×10^{-2}	8.883×10^{-1} / 9.433×10^{-1}
	c	-1.974×10^{-1}	8.975×10^{-3}	-2.150×10^{-1} / -1.798×10^{-1}
	d	-4.809×10^{-4}	1.469×10^{-4}	-7.689×10^{-4} / -1.930×10^{-4}
	e	9.362×10^{-3}	1.688×10^{-4}	9.031×10^{-3} / 9.693×10^{-3}
	f	9.757×10^{-4}	5.574×10^{-5}	8.664×10^{-4} / 1.085×10^{-3}

Table C.35. Comparison of models of ALU system cooling energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	28.54	27.55	28.54	28.54	28.54	28.54	28.54	28.54	28.54
σ_y	15.655	17.039	15.434	15.470	15.476	15.463	15.584	15.589	15.589
RSS		40074	26394	22069	21340	22893	8506	7899	7877
R^2		0.9574	0.9719	0.9765	0.9773	0.9757	0.991	0.9916	0.9916
RMSD		3.2305	2.6217	2.3973	2.3574	2.4417	1.4883	1.4342	1.4322
e_{max}		13.17	12.78	11.73	11.72	10.67	7.95	7.33	7.68
e_{min}		-12.70	-8.45	-6.89	-6.49	-9.25	-6.10	-6.17	-6.07
$\overline{\lvert e \rvert}$		2.37	1.92	1.73	1.68	1.83	1.14	1.10	1.09
$\lvert e \rvert_{95\%}$		7.13	5.38	4.95	4.80	4.90	3.12	2.91	2.93

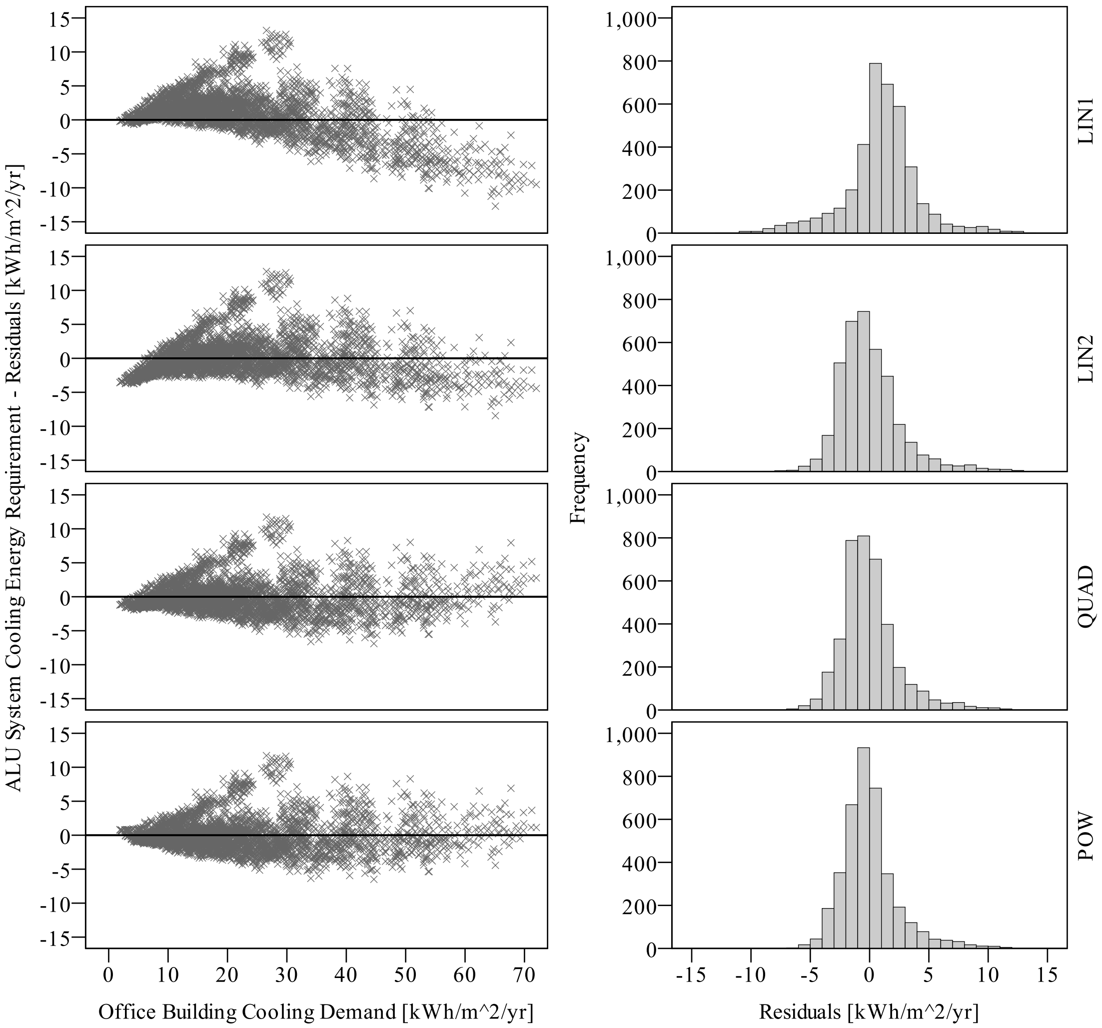


Figure C.25. Residuals scatter plots and histograms for single independent variable models of ALU system cooling energy consumption in office buildings

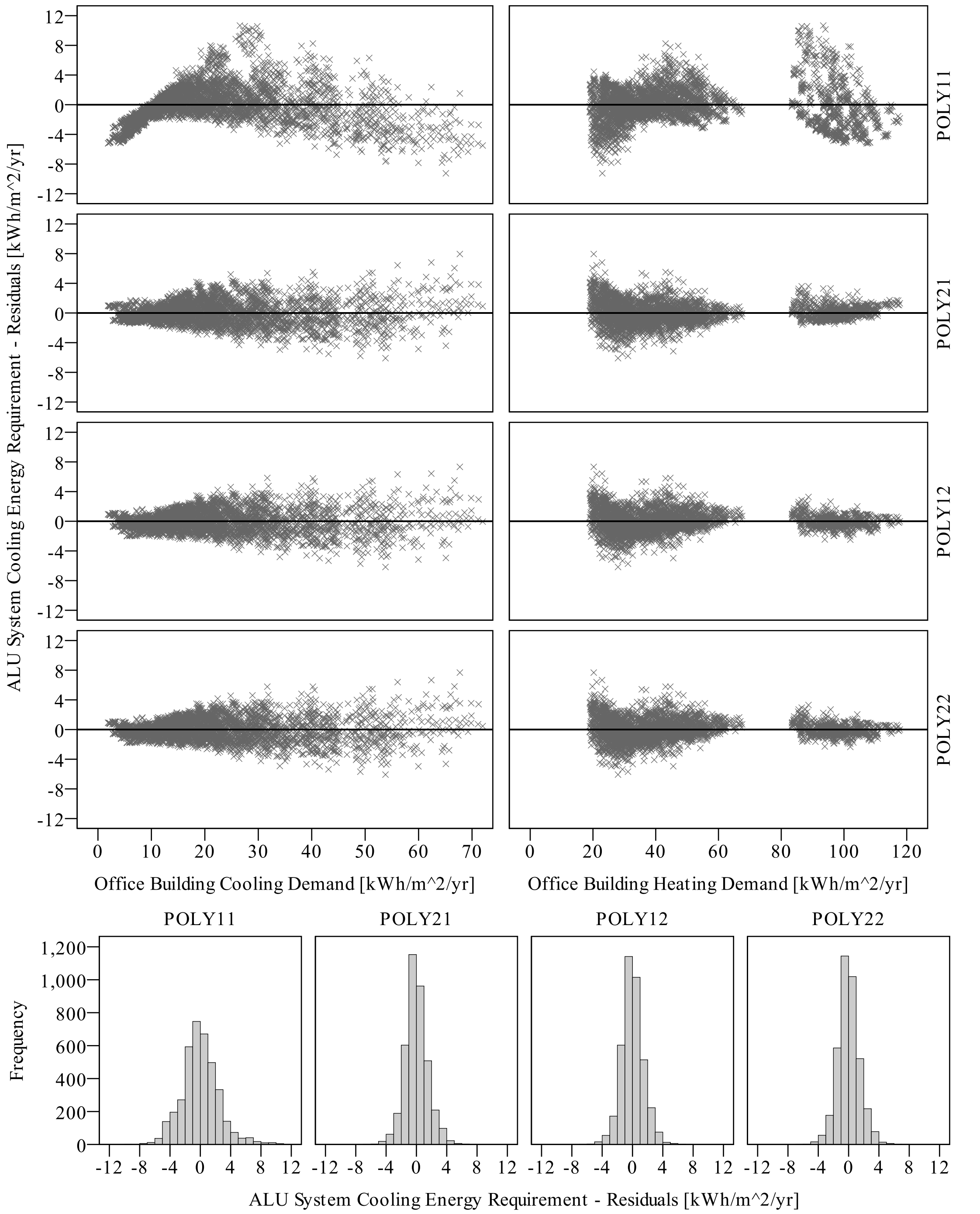


Figure C.26. Residuals scatter plots and histograms for two independent variables models of ALU system cooling energy consumption in office buildings

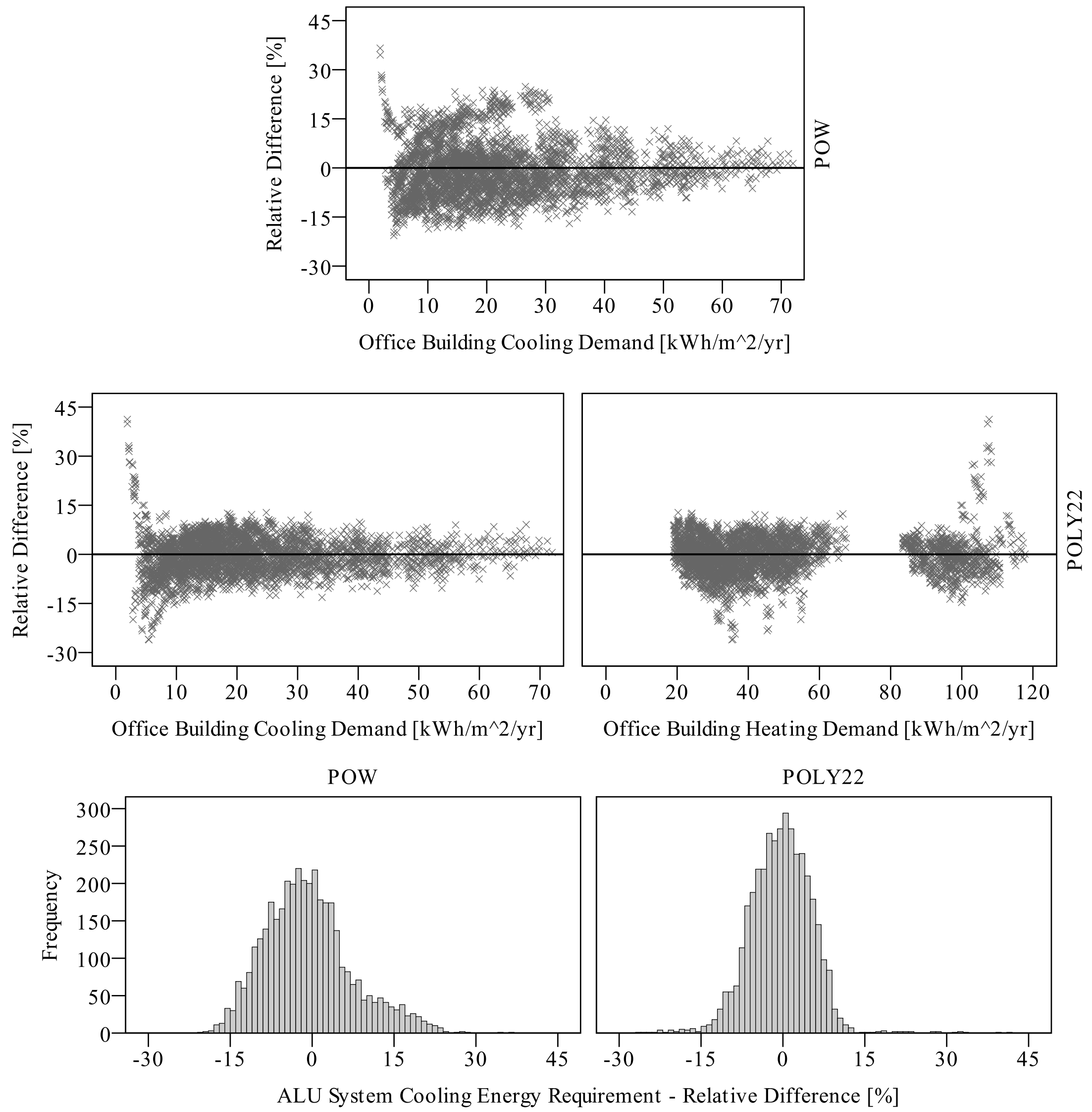


Figure C.27. Residuals scatter plots and histograms for POW and POLY22 models of ALU system cooling energy consumption in office buildings

Table C.36. Comparison of relative differences between predicted and observed values for models of ALU system cooling energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	5.73	-3.34	-1.36	-0.68	-3.96	-0.48	-0.42	-0.36
Std. Dev.	9.252	15.247	9.095	8.012	18.806	6.129	5.868	5.685
Maximum	31.33	27.12	24.88	36.59	22.64	44.73	42.47	41.22
Minimum	-18.01	-161.89	-55.65	-20.68	-235.08	-34.25	-29.29	-26.10
Perc. 25	-0.92	-7.22	-6.79	-6.31	-5.99	-4.30	-4.08	-4.02
Perc. 75	11.65	4.24	3.67	3.67	4.98	3.15	3.39	3.37

- **Heating energy requirements**

Table C.37. Regression model parameter values, standard errors and 95% confidence bounds for models of ALU system heating energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
LIN1	b	5.738×10^{-1}	9.660×10^{-4}	5.719×10^{-1} / 5.757×10^{-1}
LIN2	a	-4.83	8.217×10^{-2}	-4.991 / -4.669
	b	6.504×10^{-1}	1.480×10^{-3}	6.475×10^{-1} / 6.533×10^{-1}
QUAD	a	-1.47	2.261×10^{-1}	-1.914 / -1.027
	b	5.138×10^{-1}	8.724×10^{-3}	4.967×10^{-1} / 5.309×10^{-1}
	c	1.077×10^{-3}	6.785×10^{-5}	9.441×10^{-4} / 1.210×10^{-3}
POW	a	1.474×10^{-1}	3.018×10^{-1}	-4.443×10^{-1} / 7.390×10^{-1}
	b	2.717×10^{-1}	1.635×10^{-2}	2.396×10^{-1} / 3.037×10^{-1}
	c	1.174	1.211×10^{-2}	1.150 / 1.198
POLY11	a	-7.283	1.353×10^{-1}	-7.548 / -7.017
	b	7.119×10^{-2}	3.222×10^{-3}	6.487×10^{-2} / 7.751×10^{-2}
	c	6.692×10^{-1}	1.633×10^{-3}	6.660×10^{-1} / 6.724×10^{-1}
POLY21	a	-7.743	2.865×10^{-1}	-8.305 / -7.181
	b	1.057×10^{-1}	1.899×10^{-2}	6.847×10^{-2} / 1.429×10^{-1}
	c	6.768×10^{-1}	3.477×10^{-3}	6.700×10^{-1} / 6.836×10^{-1}
	d	-2.670×10^{-4}	2.185×10^{-4}	-6.954×10^{-4} / 1.615×10^{-4}
	e	-5.569×10^{-4}	2.111×10^{-4}	-9.708×10^{-4} / -1.430×10^{-4}
POLY12	a	-3.735	3.703×10^{-1}	-4.461 / -3.009
	b	3.377×10^{-2}	7.536×10^{-3}	1.900×10^{-2} / 4.855×10^{-2}
	c	5.432×10^{-1}	1.221×10^{-2}	5.192×10^{-1} / 5.671×10^{-1}
	e	7.638×10^{-4}	1.898×10^{-4}	3.918×10^{-4} / 1.136×10^{-3}
	f	8.934×10^{-4}	8.167×10^{-5}	7.333×10^{-4} / 1.054×10^{-3}
POLY22	a	-2.72	5.297×10^{-1}	-3.758 / -1.681
	b	-2.119×10^{-2}	2.185×10^{-2}	-6.404×10^{-2} / 2.165×10^{-2}
	c	5.248×10^{-1}	1.399×10^{-2}	4.974×10^{-1} / 5.523×10^{-1}
	d	6.135×10^{-4}	2.290×10^{-4}	1.646×10^{-4} / 1.062×10^{-3}
	e	1.253×10^{-3}	2.632×10^{-4}	7.369×10^{-4} / 1.769×10^{-3}
	f	9.733×10^{-4}	8.688×10^{-5}	8.030×10^{-4} / 1.144×10^{-3}

Table C.38. Comparison of models of ALU system heating energy consumption in office buildings

	Observed	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
$\bar{y}$	26.98	28.06	26.98	26.98	26.98	26.98	26.98	26.98	26.98
σ_y	17.278	15.093	17.109	17.119	17.118	17.128	17.128	17.133	17.133
RSS		42422	22322	20946	21164	19803	19765	19174	19138
R^2		0.963	0.9805	0.9817	0.9815	0.9827	0.9828	0.9833	0.9833
RMSD		3.3238	2.411	2.3355	2.3476	2.2709	2.2687	2.2346	2.2325
e_{max}		13.09	8.90	6.84	7.26	8.65	8.44	7.21	7.20
e_{min}		-5.21	-6.06	-5.62	-5.78	-5.93	-5.93	-5.48	-5.44
$\overline{\lvert e \rvert}$		2.73	1.87	1.80	1.81	1.74	1.74	1.71	1.71
$\lvert e \rvert_{95\%}$		5.57	5.07	5.19	5.23	4.86	4.91	4.90	4.81

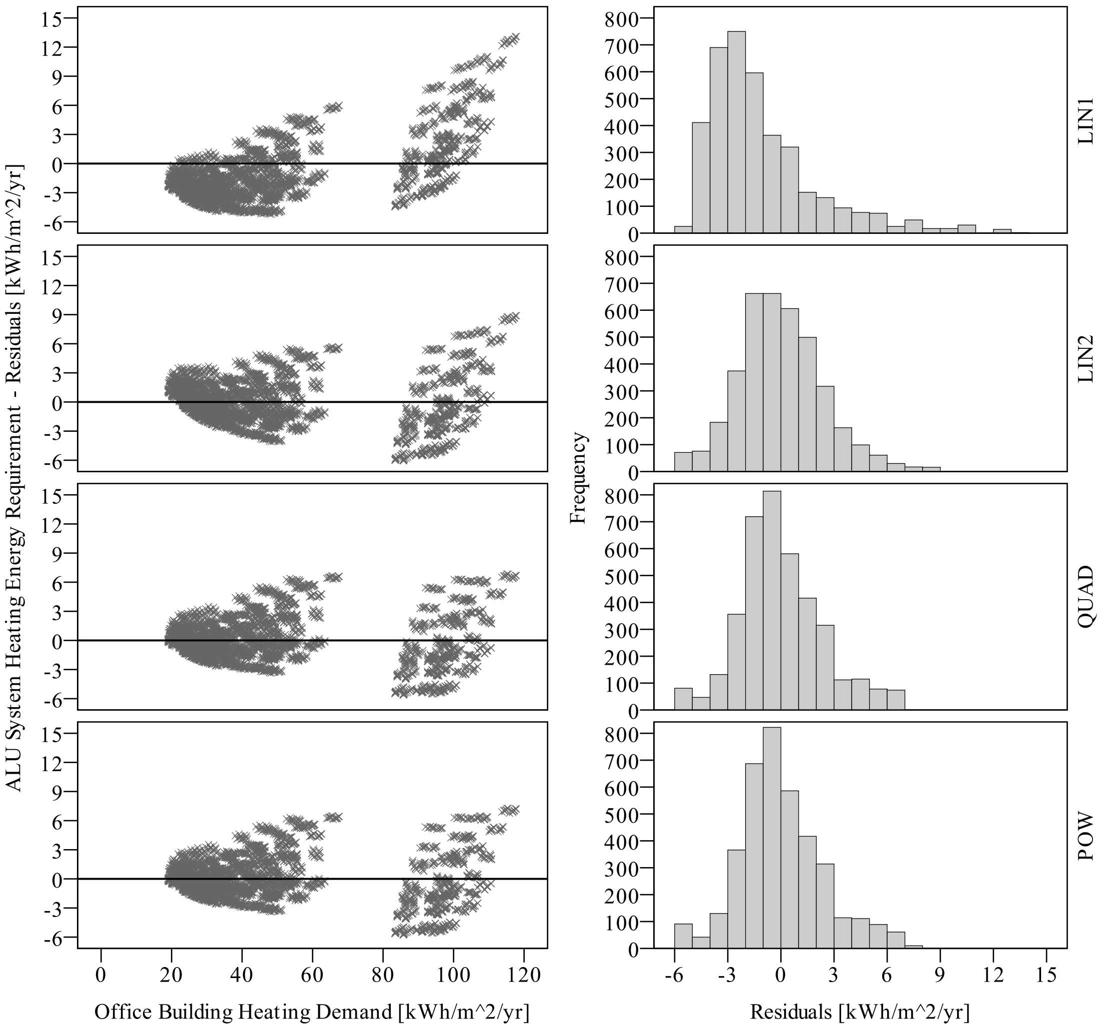


Figure C.28. Residuals scatter plots and histograms for single independent variable models of ALU system heating energy consumption in office buildings

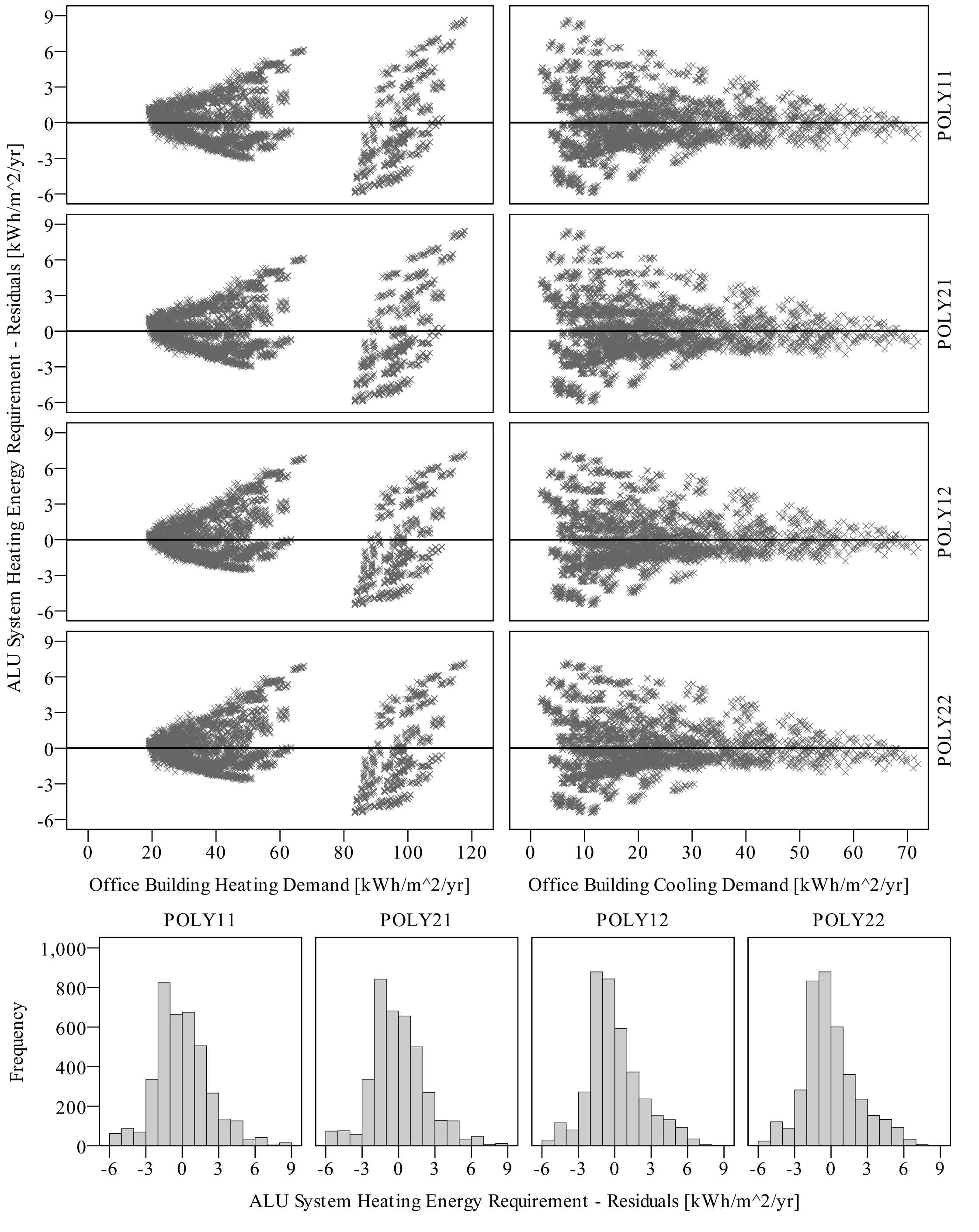


Figure C.29. Residuals scatter plots and histograms for two independent variables models of ALU system heating energy consumption in office buildings

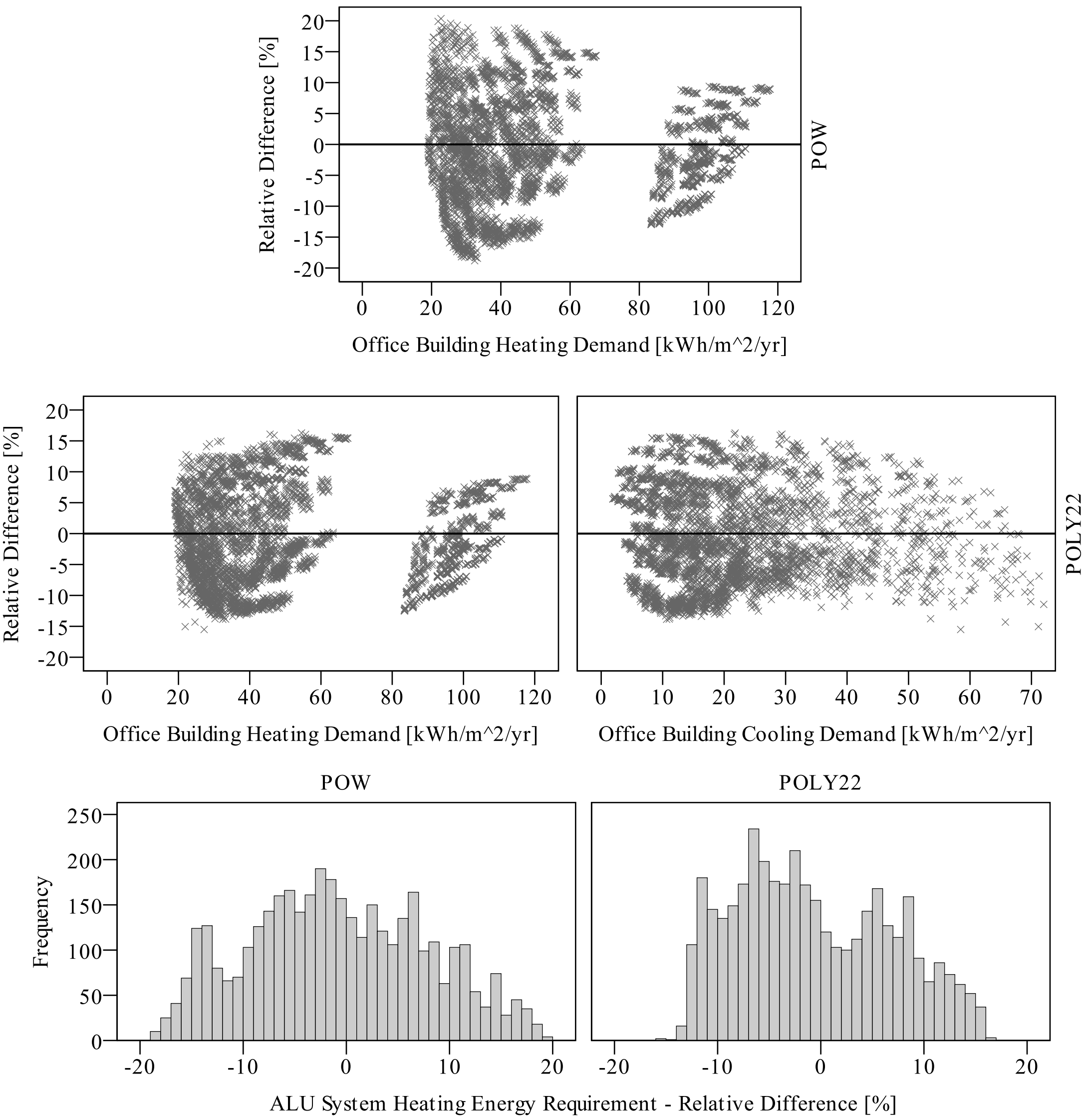


Figure C.30. Residuals scatter plots and histograms for POW and POLY22 models of ALU system heating energy consumption in office buildings

Table C.39. Comparison of relative differences between predicted and observed values for models of ALU system heating energy consumption in office buildings

	LIN1	LIN2	QUAD	POW	POLY11	POLY21	POLY12	POLY22
Mean	-10.19	-0.20	-0.84	-0.81	-0.31	-0.37	-0.72	-0.72
Std. Dev.	13.154	9.324	8.707	8.662	7.654	7.598	7.492	7.515
Maximum	16.23	26.98	20.45	20.36	16.65	15.96	16.07	16.28
Minimum	-38.04	-18.70	-19.14	-18.82	-15.17	-14.92	-13.71	-15.52
Perc. 25	-20.08	-6.95	-7.16	-7.16	-6.91	-6.95	-6.93	-6.70
Perc. 75	0.42	6.11	5.84	5.88	6.68	6.38	5.70	5.41

- **Auxiliary energy requirements**

Table C.40. Regression model parameter values, standard errors and 95% confidence bounds for models of ALU system auxiliary energy consumption in office buildings

Model	Parameter	Value	Std. Error	95% Confidence bounds
POLY11	a	4.394	3.231×10^{-2}	4.331 / 4.457
	b	1.089×10^{-1}	7.694×10^{-4}	1.074×10^{-1} / 1.104×10^{-1}
	c	1.824×10^{-2}	3.898×10^{-4}	1.748×10^{-2} / 1.901×10^{-2}
POLY21	a	3.849	6.174×10^{-2}	3.728 / 3.970
	b	1.481×10^{-1}	4.092×10^{-3}	1.401×10^{-1} / 1.561×10^{-1}
	c	1.676×10^{-2}	7.492×10^{-4}	1.529×10^{-2} / 1.823×10^{-2}
	d	-8.044×10^{-4}	4.709×10^{-5}	-8.967×10^{-4} / -7.120×10^{-4}
	e	3.326×10^{-4}	4.549×10^{-5}	2.434×10^{-4} / 4.218×10^{-4}
POLY12	a	5.118	8.379×10^{-2}	4.954 / 5.282
	b	7.685×10^{-2}	1.705×10^{-3}	7.350×10^{-2} / 8.019×10^{-2}
	c	-5.096×10^{-3}	2.763×10^{-3}	-1.051×10^{-2} / 3.206×10^{-4}
	e	9.544×10^{-4}	4.294×10^{-5}	8.702×10^{-4} / 1.039×10^{-3}
	f	8.972×10^{-5}	1.848×10^{-5}	5.349×10^{-5} / 1.259×10^{-4}
POLY22	a	3.761	1.160×10^{-1}	3.533 / 3.988
	b	1.503×10^{-1}	4.785×10^{-3}	1.409×10^{-1} / 1.597×10^{-1}
	c	1.942×10^{-2}	3.063×10^{-3}	1.342×10^{-2} / 2.543×10^{-2}
	d	-8.198×10^{-4}	5.014×10^{-5}	-9.181×10^{-4} / -7.215×10^{-4}
	e	3.009×10^{-4}	5.763×10^{-5}	1.879×10^{-4} / 4.139×10^{-4}
	f	-1.706×10^{-5}	1.902×10^{-5}	-5.436×10^{-5} / 2.024×10^{-5}

Table C.41. Comparison of models of ALU system auxiliary energy consumption in office buildings

	Observed	POLY11	POLY21	POLY12	POLY22
$\overline{y}$	7.63	7.63	7.63	7.63	7.63
σ_y	1.380	1.269	1.291	1.284	1.291
RSS		1129	918	982	918
R^2		0.8457	0.8745	0.8658	0.8745
RMSD		0.5422	0.4889	0.5057	0.4889
e_{max}		1.19	1.36	0.83	1.36
e_{min}		-1.98	-1.52	-1.54	-1.52
$\overline{\lvert e \rvert}$		0.46	0.40	0.43	0.40
$\lvert e \rvert_{95\%}$		0.99	0.89	0.93	0.90

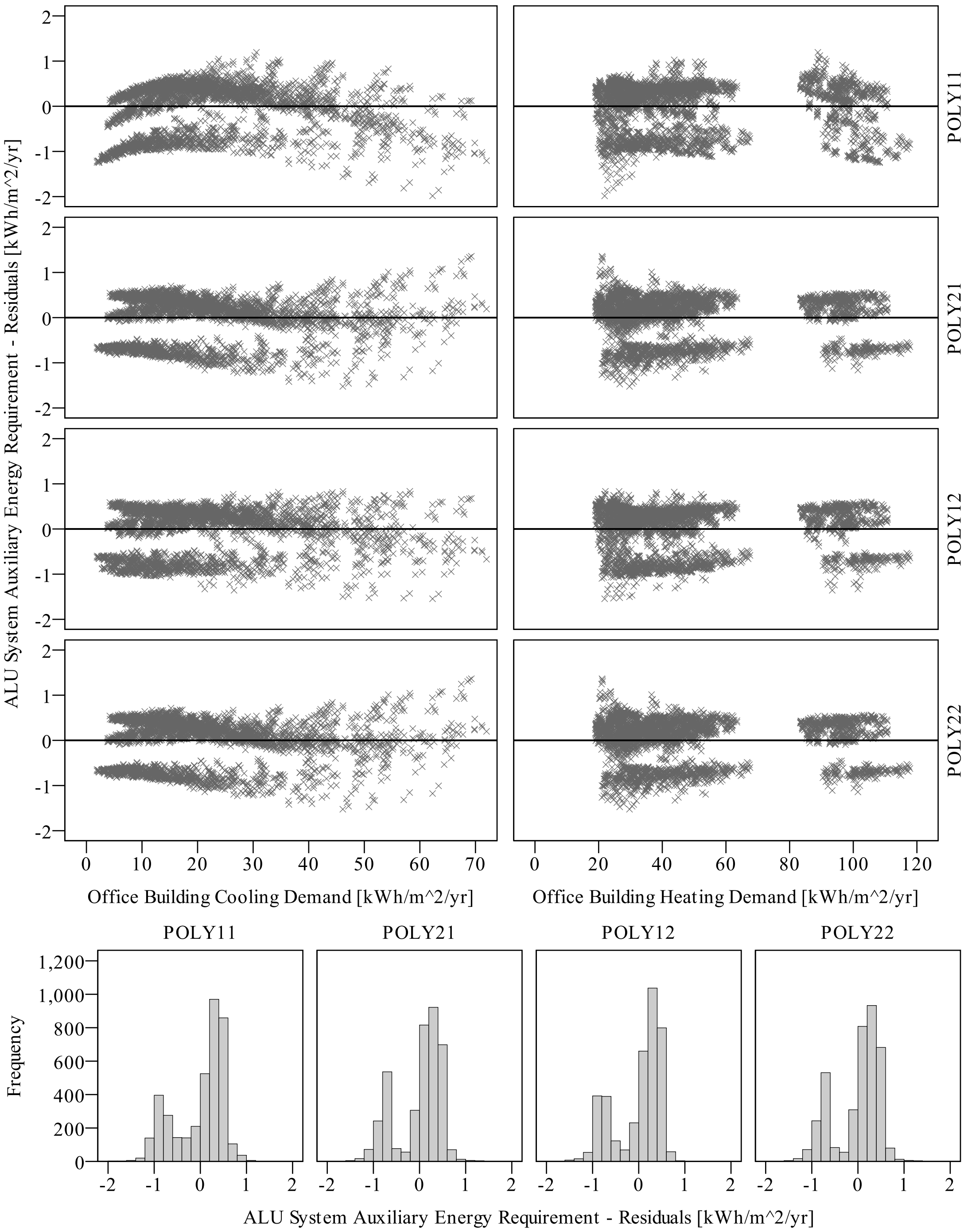


Figure C.31. Residuals scatter plots and histograms for two independent variables models of ALU system auxiliary energy consumption in office buildings

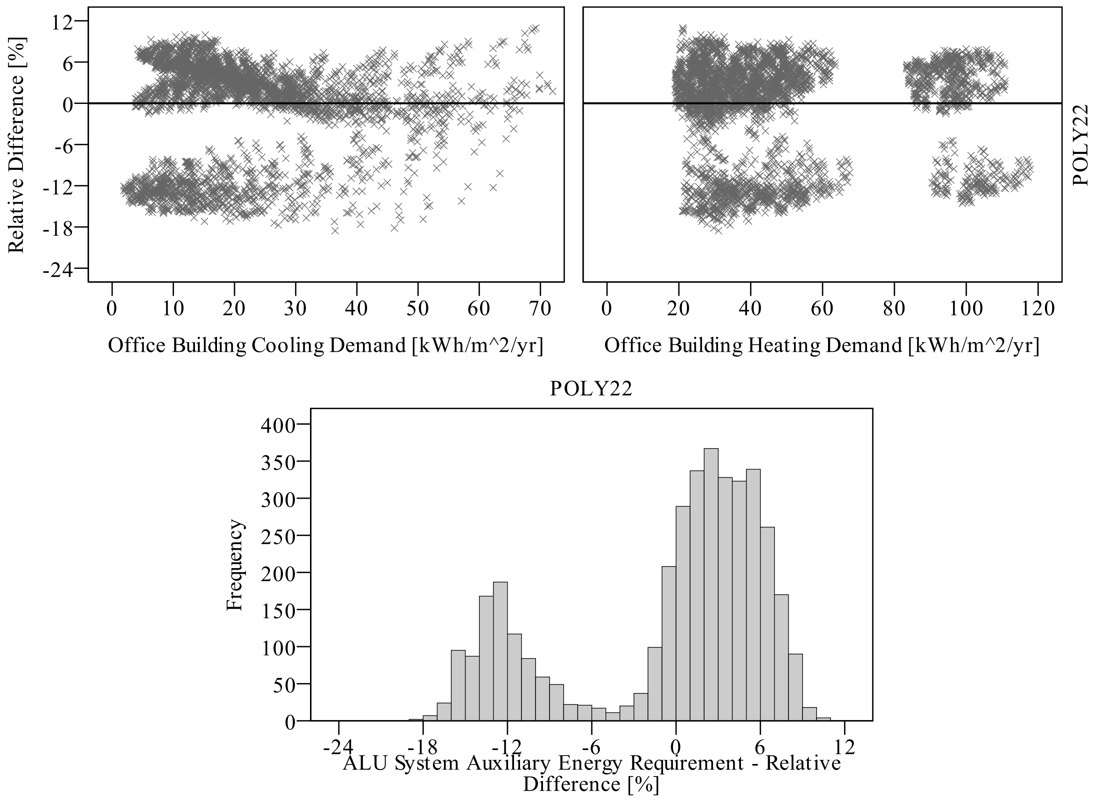


Figure C.32. Residuals scatter plots and histogram for POLY22 model of ALU system auxiliary energy consumption in office buildings

Table C.42. Comparison of relative differences between predicted and observed values for models of ALU system auxiliary energy consumption in office buildings

	POLY11	POLY21	POLY12	POLY22
Mean	-0.69	-0.51	-0.59	-0.51
Std. Dev.	8.125	7.218	7.615	7.215
Maximum	11.32	10.92	8.61	10.96
Minimum	-24.36	-18.50	-21.26	-18.54
Perc. 25	-5.64	-3.24	-4.85	-3.26
Perc. 75	5.27	4.78	5.09	4.76